OIPE JC117 SEP 0 2 2003 PATENT & TRADEMARK

AF 3683 $ #13 9/12/03

Please type a plus sign (+) inside this box → [+]

HDP/SB/21 based on PTO/SB/21 (08-00)

TRANSMITTAL FORM

(to be used for all correspondence after initial filing)

Application Number	09/~~080~~,054 990
Filing Date	November 21, 2001
First Named Inventor	Dodge et al.
Group Art Unit	3683
Examiner Name	Melody M. Burch
Attorney Docket Number	1316N-001660

Total Number of Pages in This Submission

ENCLOSURES *(check all that apply)*

- [x] Fee Transmittal Form
  - [x] Fee Attached
- [ ] Amendment / Response
  - [ ] After Final
  - [ ] Affidavits/declaration(s)
- [ ] Extension of Time Request
- [ ] Express Abandonment Request
- [ ] Information Disclosure Statement
- [ ] Certified Copy of Priority Document(s)
- [ ] Response to Missing Parts/ Incomplete Application
  - [ ] Response to Missing Parts under 37 CFR 1.52 or 1.53
- [ ] Assignment Papers *(for an Application)*
- [ ] Drawing(s)
- [ ] Licensing-related Papers
- [ ] Petition
- [ ] Petition to Convert to a Provisional Application
- [ ] Power of Attorney, Revocation Change of Correspondence Address
- [ ] Terminal Disclaimer
- [ ] Request for Refund
- [ ] CD, Number of CD(s) _____
- [ ] After Allowance Communication to Group
- [ ] Appeal Communication to Board of Appeals and Interferences
- [x] Appeal Communication to Group *(Appeal Notice, Brief, Reply Brief)*
- [ ] Proprietary Information
- [ ] Status Letter
- [ ] Other Enclosure(s) *(please identify below)*:

Remarks: **The Commissioner is hereby authorized to charge any additional fees that may be required under 37 CFR 1.16 or 1.17 to Deposit Account No. 08-0750.**

RECEIVED SEP 0 9 2003 GROUP 3600

SIGNATURE OF APPLICANT, ATTORNEY, OR AGENT

Firm *or* Individual name	Harness, Dickey & Pierce, P.L.C.	Attorney Name Michael J. Schmidt	Reg. No. 34,007
Signature			
Date	August 27, 2003		

CERTIFICATE OF MAILING/TRANSMISSION

I hereby certify that this correspondence is being deposited with the United States Postal Service as first class mail in an envelope addressed to: Director of the U.S. Patent and Trademark Office, P.O. Box 1450, Alexandria, VA 22313-1450, or facsimile transmitted to the U.S. Patent and Trademark Office on the date indicated below.

Typed or printed name	Michael J. Schmidt		
Signature		Date	August 27, 2003

OIPE SEP 0 2 2003 PATENT & TRADEMARK OFFICE

PTO/SB/17 (01-03)
Approved for use through 10/31/2002. OMB 0651-0032
U.S. Patent and Trademark Office; U.S. DEPARTMENT OF COMMERCE
Under the Paperwork Reduction Act of 1995, no persons are required to respond to a collection of information unless it displays a valid OMB control number.

FEE TRANSMITTAL for FY 2003

Patent fees are subject to annual revision.

☐ Applicant claims small entity status. See 37 CFR 1.27

TOTAL AMOUNT OF PAYMENT ($) 320

Complete if Known	
Application Number	09/990,054
Filing Date	November 21, 2001
First Named Inventor	Dodge et al.
Examiner Name	Melody M. Burch
Group / Art Unit	3683
Attorney Docket No.	1316N-001660

METHOD OF PAYMENT (check all that apply)

☐ Check ☐ Credit card ☐ Money Order ☐ Other ☐ None

☐ Deposit Account:

Deposit Account Number: 08-0750

Deposit Account Name: Harness, Dickey & Pierce, P.L.C.

The Commissioner is authorized to: *(check all that apply)*

☐ Charge fee(s) indicated below ☒ Credit any overpayments

☒ Charge any additional fee(s) during the pendency of this application

☐ Charge fee(s) indicated below, **except for the filing fee** to the above-identified deposit account.

FEE CALCULATION

1. BASIC FILING FEE

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description	Fee Paid
1001	750	2001	375	Utility filing fee	
1002	330	2002	165	Design filing fee	
1003	520	2003	260	Plant filing fee	
1004	750	2004	375	Reissue filing fee	
1005	160	2005	80	Provisional filling fee	

SUBTOTAL (1) ($) 0

2. EXTRA CLAIM FEES

			Extra Claims		Fee from below		Fee Paid
Total Claims		-20 ** =	0	X		=	0
Independent Claims		-3 ** =	0	X		=	0
Multiple Dependent				X		=	0

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description
1202	18	2202	9	Claims in excess of 20
1201	84	2201	42	Independent claims in excess of 3
1203	280	2203	140	Multiple dependent claim, if not paid
1204	84	2204	42	** Reissue independent claims over original patent
1205	18	2205	9	** Reissue claims in excess of 20 and over original patent

SUBTOTAL (2) ($) 0

**or number previously paid, if greater; For Reissues, see above

FEE CALCULATION (continued)

3. ADDITIONAL FEES

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description	Fee Paid
1051	130	2051	65	Surcharge - late filing fee or oath	
1052	50	2052	25	Surcharge - late provisional filing fee or cover sheet.	
1053	130	1053	130	Non-English specification	
1812	2,520	1812	2,520	For filing a request for reexamination	
1804	920*	1804	920*	Requesting publication of SIR prior to Examiner action	
1805	1,840*	1805	1,840*	Requesting publication of SIR after Examiner action	
1251	110	2251	55	Extension for reply within first month	
1252	410	2252	205	Extension for reply within second month	
1253	930	2253	465	Extension for reply within third month	
1254	1,450	2254	725	Extension for reply within fourth month	
1255	1,970	2255	985	Extension for reply within fifth month	
1401	320	2401	160	Notice of Appeal	
1402	320	2402	160	Filing a brief in support of an appeal	320
1403	280	2403	140	Request for oral hearing	
1451	1,510	1451	1,510	Petition to institute a public use proceeding	
1452	110	2452	55	Petition to revive – unavoidable	
1453	1,300	2453	650	Petition to revive – unintentional	
1501	1,300	2501	650	Utility issue fee (or reissue)	
1502	470	2502	235	Design issue fee	
1503	630	2503	315	Plant issue fee	
1460	130	1460	130	Petitions to the Commissioner	
1807	50	1807	50	Processing fee under 37 CFR 1.17 (q)	
1806	180	1806	180	Submission of Information Disclosure Stmt	
8021	40	8021	40	Recording each patent assignment per property (times number of properties)	
1809	750	2809	375	Filing a submission after final rejection (37 CFR § 1.129(a))	
1810	750	2810	375	For each additional invention to be examined (37 CFR § 1.129(b))	
1801	750	2801	375	Request for Continued Examination (RCE)	
1802	900	1802	900	Request for expedited examination of a design application	

Other fee (specify) ____

*Reduced by Basic Filing Fee Paid **SUBTOTAL (3)** ($) 320

RECEIVED SEP 0 9 2003 GROUP 36

SUBMITTED BY				Complete *(if applicable)*	
Name (Print/Type)	Michael J. Schmidt	*Registration No. Attorney/Agent*	34,007	*Telephone*	248.641.1600
Signature				*Date*	August 27, 2003

WARNING: Information on this form may become public. Credit card information should not be included on this form. Provide credit card information and authorization on PTO-2038.

PATENT

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Serial No.:	09/~~090~~,054 990
Filing Date:	November 21, 2001
Appellant:	Dodge et al.
Group Art Unit:	3683
Examiner:	Melody M. Burch
Title:	Clip Disc
Attorney Docket:	1316N-001660

Director of the United States Patent and Trademark Office
P.O. Box 1450
Alexandria, VA 22313-1450

Appellant's Brief Under 37 C.F.R. §1.192

Dear Sir:

This is an appeal from the March 24, 2003 final rejection of Claims 1-6 of the above referenced patent application. None of the claims have been allowed. No claims have been cancelled.

The Final Office Action disapproved a drawing correction because the Examiner felt it introduced new matter to the drawings; it objected to the specification for failing to provide antecedent basis for the claimed subject matter; it objected to the specification under 35 U.S.C. §112; it rejected Claim 5 under 35 U.S.C. §112; it rejected Claims 1-3 under 35 U.S.C. §112; it rejected Claims 1-5 under 35 U.S.C. §103(a) as being unpatentable over U.S. Patent No. 5,042,624 to Furuya et al. in view of U.S. Patent No. 5,529,154 to Tanaka and U.S. Patent No. 4,964,493 to Yamaura et al.; and it rejected Claim 6 under 35 U.S.C. §103 as being unpatentable over Yamaura et al. in view of Tanaka. Claims 1-6, the claims on appeal, are

reproduced in Appendix A. A copy of the 35 U.S.C. §103(a) references are reproduced in Appendix B.

Real Party in Interest

Tenneco Automotive Inc. is the real party in interest, being the Assignee of the present application on an assignment recorded by the U.S. Patent and Trademark Office on Reel 012318, Frame 0867.

Related Appeals and Interferences

To the best of Applicant's knowledge, no other appeals or interferences are pending which will directly affect, be directly affected by or have a bearing on the Board's decision in the present pending appeal.

Status of the Claims

Claims 1-5 are rejected under 35 U.S.C. §103(a) as being unpatentable over U.S. Patent No. 5,042,624 to Furuya et al. in view of U.S. Patent No. 5,529,154 to Tanaka and U.S. Patent No. 4,964,493 to Yamaura et al.

Claim 6 is rejected under 35 U.S.C. §103(a) as being unpatentable over Yamaura et al. in view of Tanaka.

Status of Amendments

Applicants mailed an Amendment After Final on May 14, 2003 which amended Claims 1 and 6. In an Advisory Action mailed June 3, 2003, the Examiner indicated that the May 14, 2003 Amendment would not be entered for purposes of Appeal. Applicants mailed an Amendment in response to the June 3, 2003 Advisory Action on June 24, 2003 again amending Claims 1 and 6. The Notice of Appeal was also filed on June 24, 2003. In a second Advisory

Action mailed July 21, 2003, the Examiner indicated that the June 24, 2003 amendment would be entered for purposes of Appeal and that the Amendment overcame the 35 U.S.C. §112 rejection of Claim 6, the lack of antecedent basis specification objection and the new mater drawing objection. Thus, applicants believe that the 35 U.S.C. §103(a) rejections are the only remaining rejections and this Appeal is directed towards those rejections.

Summary of the Invention

Referring primarily to Figures 2 and 4, the present invention in Claim 1 relates to a shock absorber for a vehicle. The shock absorber includes a piston and a base valve assembly where the base valve assembly includes a variable orifice bleed circuit which is incorporated into the secondary valving system on the shock absorber. The secondary valving system includes a plurality of valve discs secured to the base valve assembly to close the fluid passages extending through the base valve assembly. The plurality of discs deflects due to a pressure differential to open the fluid passages.

The variable orifice bleed circuit of the present invention incorporates a clipped disc disposes directly adjacent the main valve disc with the main valve disc resting directly on the valve body of the base valve assembly. The clipped valve disc has an outer edge defined by an outer circular edge that which is truncated by a single chordal edge. The angle outer chordal edge allows an outer circumferential portion of the main valve disk in contact with the piston to deflect along the single outer chordal edge prior to the deflection of the entire stack of valve discs to provide for the variable orifice bleed circuit.

Claim 6 is similar to Claim 4, but the secondary valving system and the variable orifice bleed circuit are part of a piston assembly instead of a base valve assembly.

Issues

Appellants present the following issues for review:

1) Whether or not Claims 1-5 are unpatentable under 35 U.S.C. §103(a) over U.S. Patent No. 5,042,624 to Furuya et al. in view of U.S. Patent No. 5,529,154 to Tanaka and U.S. Patent No. 4,964,493 to Yamaura et al.

2) Whether or not Claim 6 is unpatentable under 35 U.S.C. §103(a) over Yamaura et al. in view of Tanaka.

Grouping of the Claims

Claims 1-6 stand or fall together.

Argument

Regarding Claims 1-5, the Examiner uses Furuya et al. to show a damper having most of the features of Claims 1 and 4 including a first valve disc 4e disposed adjacent the valve body and a second valve disc 4d disposed adjacent the first valve disc. The Examiner then goes on to say that Furuya et al. does not disclose that the outer edge of the second valve disc is chordal.

The Examiner then looks to Tanaka to find a second valve disc 19 which is disposed adjacent a first valve disc 17 where the second valve disc has an outer edge formed by a recess 19a which supports the first valve disc at a position between the outside edge and the central axis of the first valve disc 17. But Tanaka defines a recess 19a and not a chordal edge.

The Examiner then looks to Yamaura et al. to find a second valve disc 144 which is disposed adjacent to a first valve disc 138 <u>via element 142</u>. Thus, the Examiner admits that valve disc 144 is not adjacent to valve disc 138. It is spaced from it by disc 142.

First of all, the Examiner's description of Furuya et al. is not quite correct. Furuya does not disclose <u>a first valve disc 4e</u>; it discloses <u>a first disc valve 4e</u> and the first disc valve 4e

includes a plurality of valve discs not individually numbered. (See Figure 3.) Thus, the second valve disc 4d is not disposed adjacent the first valve disc which closes the passages, there is another valve disc disposed between them.

Second, the Examiner looks to Tanaka to show the support for the first valve disc at a position between the center of the disc and its outside edge. But Tanaka does not disclose, teach or even suggest a chordal edge. The Examiner is relying on the statement "It is also easy to grasp the correlation between the shape of the intermediate sheet 19 and the generated damping force depending on the number, shape, angle and depth of recesses 19a in the intermediate sheet 19. (Column 5, lines 46-51 of Tanaka). To find the chordal edge, the Examiner looks to Yamaura et al. which discloses a chordal edge on disc 144 in Figure 4.

As clearly illustrated in Figure 2 of Yamaura et al., the chordal edge 144 of Yamaura et al. does not support the first valve disc 138 between the center and outside edge of disc 138 such that disc 138 will deflect along the chordal edge. Disc 144 is spaced from disc 138 by disc 142 and the chordal sections in Yamaura et al. are simply used as flow passages and not as a pivot line for the first valve disc. (Yamaura et al. Column 7, lines 52-56). The Examiner recognizes this lack of disclosure in Yamaura et al. and attempts to cure this by saying that the discs are adjacent via an intervening element and that during large deflections of element 138, the second valve disc 144 has an edge that supports the first valve disc. Yamaura et al. clearly teaches away from positioning second valve disc 144 adjacent valve disc 138. Column 9, lines 20-23 specifically teach that as seen from Figure 2, the stopper plate 144 is placed away from the upper valve disc member 138 leaving clearance defined by the height of the washer 142.

To establish a *prima facie* case of obviousness, three basic criteria must be met. First, there must be some suggestion or motivation, either in the references themselves or in the knowledge generally available to one of ordinary skill in the art, to modify the reference or to combine reference teachings. Second, there must be a reasonable expectation of success. Finally, the prior art reference (or references when combined) must teach or suggest all the

claim limitations. The teaching or suggestion to make the claimed combination and the reasonable expectation of success must both be found in the prior art, and not based on applicant's disclosure. In re Vaeck, 947 F.2d 488, 20 USPQ2d 1438 (Fed. Cir. 1991). See MPEP §2143-§2143.03 for decisions pertinent to each of these criteria.

Here there is no motivation to combine Yamaura et al. with Tanaka to provide the chordal deflection edge since Yamaura et al. specifically teaches away from the Tanaka reference because it teaches to space the chordal edge away from the valve disc.

"There are three possible sources for a motivation to combine references: the nature of the problem to be solved, the teachings of the prior art, and the knowledge of persons of ordinary skill in the art." *In re Rouffet*, 149 F.3d 1350, 1357, 47 USPQ2d 1453, 1457-58 (Fed. Cir. 1998). (The combination of the references taught every element of the claimed invention, however without a motivation to combine, a rejection based on a *prima facie* case of obvious was held improper.). The level of skill in the art cannot be relied upon to provide the suggestion to combine references. Al-site Corp. v. VSI Int'l. Inc., 174 F.3d 1308, 50 USPQ2D 1161 (Fed. Cir. 1999).

Obviousness can only be established by combining or modifying the teachings of the prior art to produce the claimed invention where there is some teaching, suggestion or motivation to do so found either explicitly or implicitly in the references themselves or in the knowledge generally available to one of ordinary skill in the art. "The test for an implicit showing is what the combined teachings, knowledge of one of ordinary skill in the art, and the nature of the problem to be solved as a whole would have suggested to those of ordinary skill in the art." *In re Kotzab*, 217 F.3d 1365, 1370, 55 USPQ2d 1313, 1317 (Fed. Cir. 2000). See also In re Lee, 277 F.3d 1338, 1342-44, 61 USPQ2d 1430, 1433-34 (Fed. Cir. 2002) (discussing the importance of relying on objective evidence and making specific factual findings with respect to the motivation to combine reference); *In re Fine*, 837 F.2d 1071, 5 USPQ2d 1596 (Fed. Cir. 1988); *In re Jones*, 958 F.2d 347, 21 USPQ2D 1941 (Fed. Cir. 1992).

The mere fact that references can be combined or modified does not render the resultant combination obvious unless the prior art also suggests the desirability of the combination. *In re Mills*, 916 F.2d 680, 16 USPQ2d 1430 (Fed. Cir. 1990), claims were directed to an apparatus for producing an aerated cementitious composition by drawing air into the cementitious composition by riving the output pump at a capacity greater than the feed rate. The prior art reference taught that the feed means can be run at a variable speed, however the court found that this does not require that the output pump be run at the claimed speed so that air is drawn into the mixing chamber and is entrained in the ingredients during operation. Although a prior art device "may be capable of being modified to run the way the apparatus is claimed, there must be a suggestion or motivation in the reference to do so." 916 F.2d at 682, 16 USPQ2d at 1432). See also *In re Fritch*, 972 F.2d 1260, 23 USPQ2d 1780 (Fed. Cir. 1992) (flexible landscape edging device which is conformable to a ground surface of varying slope not suggested by combination of prior art reference).

The Examiner is relying on hindsight to arrive at the determination of obviousness. The court in *In Re Fritch*, 23 USPQ 2d 1784 (Fed. Cir. 1992), stated that it is impermissible to use the claimed invention as an instruction manual or template to piece together the teachings of the prior art so that the claimed invention is rendered obvious. This is exactly what the Examiner has done here. The Examiner has pieced together the three references to allegedly render Applicants' invention obvious, which is clearly shown due to the fact that the prior art even lacks some of the limitations of the claimed invention and the Yamaura et al. reference teaches away from the combination, as discussed above.

Regarding Claim 6, the Examiner is again using the combination of Yamaura et al. and Tanaka to reject the claim based upon obviousness under 35 U.S.C. §103. As detailed above, there is no teaching, suggestion or motivation to combine these teachings and the Examiner is relying on hindsight reconstruction to render Applicant's claims unpatentable.

Thus, Applicants believe Claims 1-6 patentable distinguish over the art of record.

Conclusion

Applicants respectfully submit that the Examiner has not proven that these reference combination (two reference combination for Figure 6) presents a *prima facie* case of obviousness as the references do not teach the elements of the claimed invention, much less suggest the combination of the references. Accordingly, reversal of the final rejection of Claims 1-6 and allowance of the claims is respectfully requested.

Respectfully submitted,

Date: August 27, 2003

By: [signature]
Michael J Schmidt
Reg. No. 34,007

HARNESS, DICKEY & PIERCE, P.L.C.
P.O. Box 828
Bloomfield Hills, Michigan 48303
(248) 641-1600
MJS/csd

APPENDIX A

Listing Of Claims

1. A damper comprising:

a pressure tube forming a working chamber;

a reservoir tube disposed around said pressure tube, said reservoir tube forming a reservoir chamber between said pressure tube and said reservoir tube;

a base valve assembly disposed between said working chamber and said reservoir chamber for regulating flow of damping fluid in a first direction between said working chamber and said reservoir chamber, said base valve assembly comprising:

a valve body defining a fluid passage;

a first valve disc disposed adjacent said valve body for closing said fluid passage, said first valve disc having a circular outside edge and a central axis;

a second valve disc disposed adjacent said first valve disc, said second valve disc having an outer edge defined by an outer circular edge truncated only by a single outer chordal edge, said single outer chordal edge supporting said first valve disc at a position between said circular outside edge and said central axis of said first valve disc.

2. The damper according to Claim 1, further comprising a piston disposed within said working chamber, said piston dividing said working chamber into an upper portion and a lower portion, said base valve assembly being

disposed between said lower portion of said working chamber and said reservoir chamber.

3. The damper according to Claim 1, wherein said base valve assembly includes a rebound valve assembly movable between a closed position and an open position, said rebound valve assembly regulating said flow of said damping fluid in a second direction between said working chamber and said reservoir chamber, said second direction being opposite to said first direction.

4. A damper comprising:

a pressure tube forming a working chamber;

a piston disposed within said working chamber, said piston dividing said working chamber into an upper working chamber and a lower working chamber;

a reservoir tube disposed around said pressure tube, said reservoir tube forming a reservoir chamber between said pressure tube and said reservoir tube;

a base valve assembly disposed between said lower working chamber and said reservoir chamber for regulating flow of damping fluid in a first direction between said lower working chamber and said reservoir chamber, said base valve assembly comprising:

a low speed valve movable between a closed position and an open position, said low speed valve including a first valve disc having an outside edge

and a central axis and a second valve disc having an outer edge defined by an outer circular edge truncated only by a single outer chordal edge, said second valve disc supporting said first valve disc along said single outer chordal edge at a position between said outside edge and said central axis of said first valve disc; and

a mid/high speed valve movable between a closed position and an open position, said mid/high speed valve comprising only said first and second valve discs.

5. The damper according to Claim 4, wherein said base valve assembly includes a pressure valve movable between a closed position and an open position, said pressure valve regulating said flow of said damping fluid in a second direction between said lower working chamber and said reservoir chamber, said second direction being opposite to said first direction.

6. A damper comprising;

a pressure tube forming a working chamber;

a piston disposed within said working chamber, said piston dividing said working chamber into an upper working chamber and a lower working chamber;

a piston valve assembly attached to said piston for regulating flow of damping fluid between said upper working chamber and said lower working chamber, said piston valve assembly comprising:

a low speed valve movable between a closed position and an open position, said low speed valve including a first valve disc having an outside edge and a central axis and a second valve disc having an outer edge defined by an outer circular edge truncated only by a single outer chordal edge, said second valve disc supporting said first valve disc along said single outer chordal edge at a position between said outside edge and said central axis of said first valve disc; and

a mid/high speed valve movable between a closed position and an open position, said mid/high speed valve including only two valve discs, said two valve discs being said first and second valve discs.

APPENDIX B

Attached hereto is a copy of each of the following United States patents:

1. U.S. Patent No. 5,042,624 to Furuya et al.;
2. U.S. Patent No. 5,529,154 to Tanaka; and
3. U.S. Patent No. 4,964,493 to Yamaura et al.

United States Patent [19]
Furuya et al.
[11] Patent Number: 5,042,624
[45] Date of Patent: Aug. 27, 1991

[54] HYDRAULIC SHOCK ABSORBER WITH PRE-LOADED VALVE FOR LINEAR VARIATION CHARACTERISTICS OF DAMPING FORCE

[75] Inventors: Takayuki Furuya; Fumiyuki Yamaoka, both of Kanagawa, Japan
[73] Assignee: Atsugi Unisia Corporation, Kanagawa, Japan

[21] Appl. No.: 413,066
[22] Filed: Sep. 27, 1989

[30] **Foreign Application Priority Data**

Sep. 29, 1988	[JP]	Japan	63-127862[U]
Sep. 29, 1988	[JP]	Japan	63-127863[U]
Sep. 29, 1988	[JP]	Japan	63-127864

[51] Int. Cl.[5] F16F 9/348
[52] U.S. Cl. 188/280; 188/282; 188/317; 188/322.14; 188/322.15; 188/322.22
[58] Field of Search 188/322.22, 322.15, 188/317, 322.14, 281, 282, 280, 322.13, 322.17, 322.18, 322.16, 320, 318

[56] **References Cited**

U.S. PATENT DOCUMENTS

2,740,500	4/1956	Brundrett et al.	188/322.15
3,134,460	5/1964	DeCarbon	188/322.15 X
4,076,276	2/1978	Wijnhoven et al.	188/322.14 X
4,768,629	9/1988	Wössner	188/322.14 X
4,782,925	11/1988	Grundei	188/322.14
4,809,828	3/1989	Nakazato	188/320 X
4,834,222	5/1989	Kato et al.	188/280
4,895,229	1/1990	Kato	188/317 X
4,905,799	3/1990	Yamaoka et al.	188/322.15

FOREIGN PATENT DOCUMENTS

833574	2/1952	Fed. Rep. of Germany .
1678593	4/1954	Fed. Rep. of Germany .
2031158	12/1971	Fed. Rep. of Germany .
2109398	9/1972	Fed. Rep. of Germany .
65878	3/1956	France .
55-82539	6/1980	Japan .
618797	2/1949	United Kingdom .
1387150	3/1975	United Kingdom .
1582586	1/1981	United Kingdom .

Primary Examiner—Douglas C. Butler
Attorney, Agent, or Firm—Ronald P. Kananen

[57] **ABSTRACT**

A hydraulic shock absorber employs first and second stage disc valves arranged in tandem fashion. The first stage disc valve is provided for acting on a relatively small pressure difference for enhanced damping characteristics at a relatively low piston stroke speed range. On the other hand, the second stage disc valve is provided for acting on a greater pressure difference for generating damping a force at a higher piston stroke speed range. The second stage disc valve is initially pre-loaded at a predetermined magnitude of load so as to set a pressure relief point of the second stage disc valve at a desired piston stroke speed for better damping characteristics of the shock absorber.

12 Claims, 7 Drawing Sheets

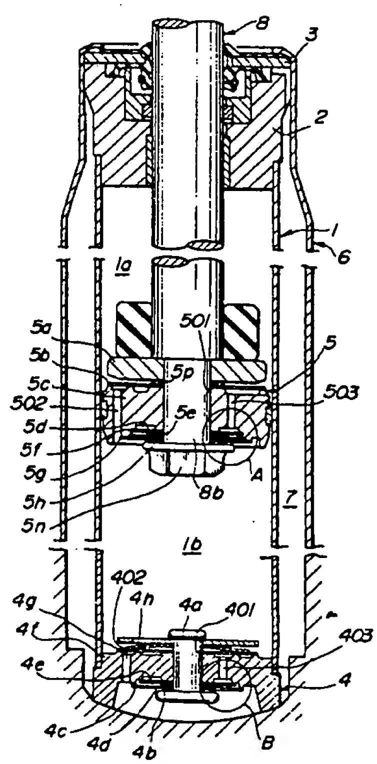


FIG.1

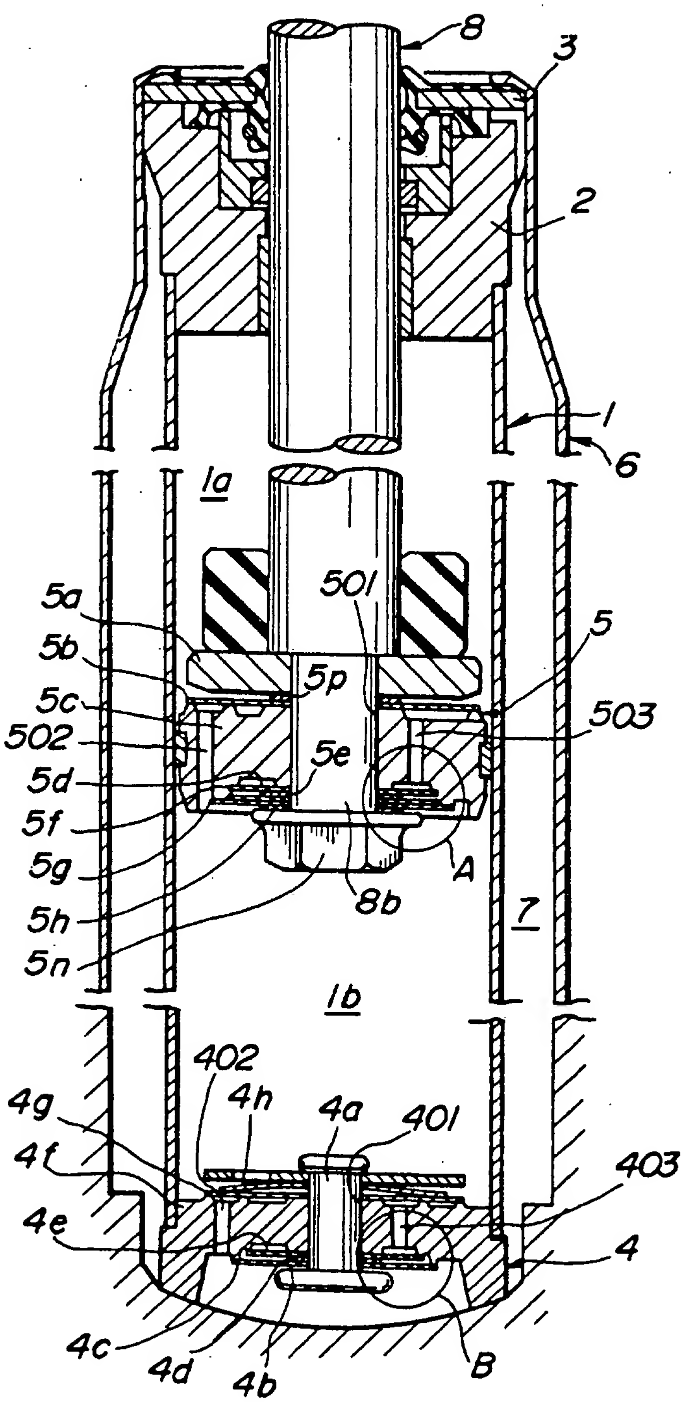

8
3
2
1
6
1a
5a
501
5
5b
5p
5c
503
502
5e
5d
5f
5g
A
5h
8b
7
5n
1b
402
4g
4h
4a
401
4f
403
4e
4
4c
4d
4b
B

FIG.2

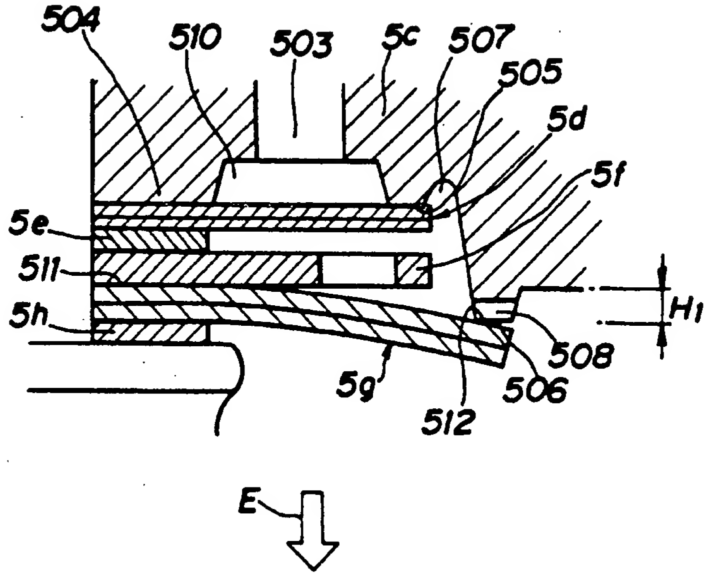

504
510
503
5c
507
505
5d
5f
5e
511
5h
H1
508
506
5g
512
E

FIG.3

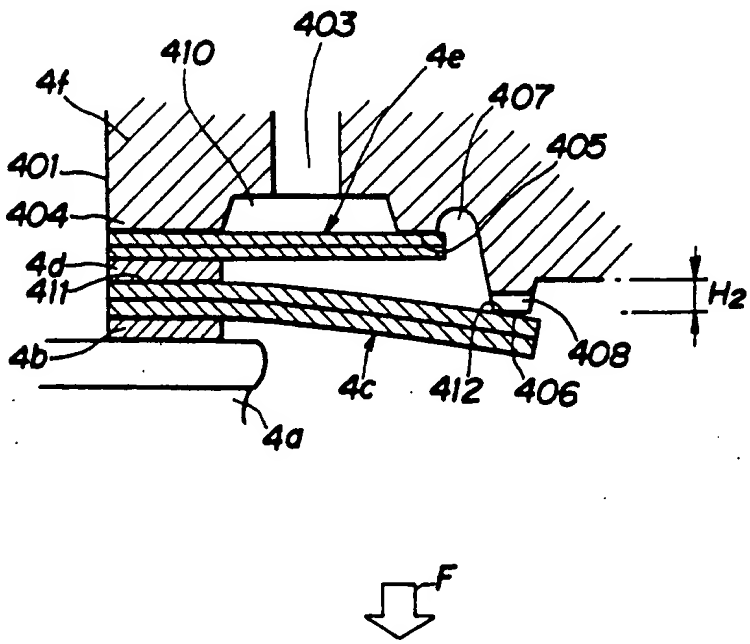

403
410
4e
4f
407
405
401
404
4d
411
H2
4b
408
4c
412
406
4a
F

FIG. 4A

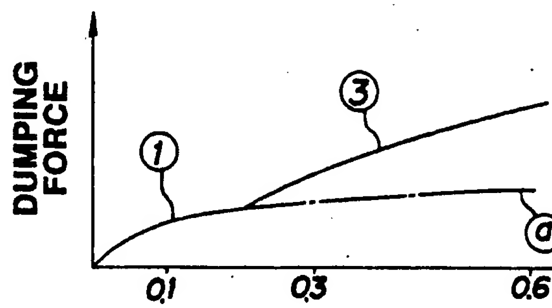


FIG. 4B

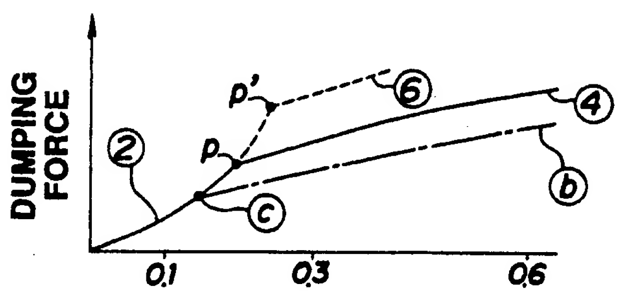


FIG. 4C

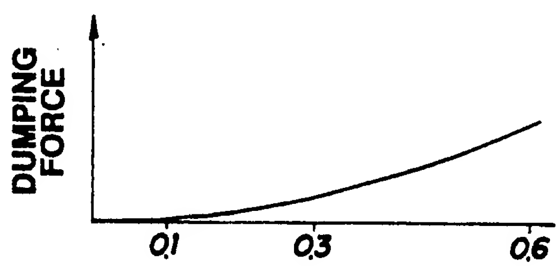


FIG. 5

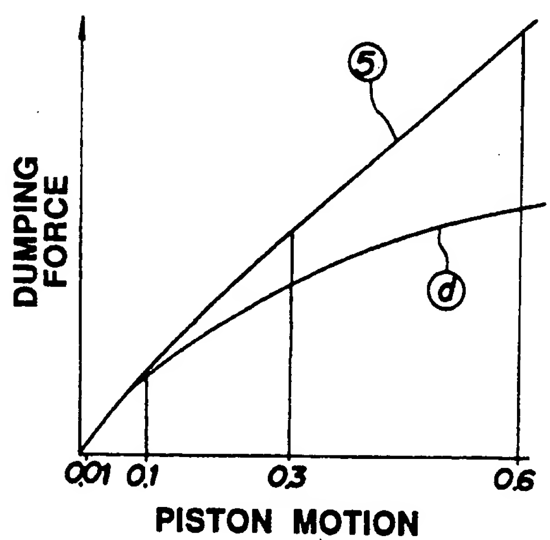


FIG. 6

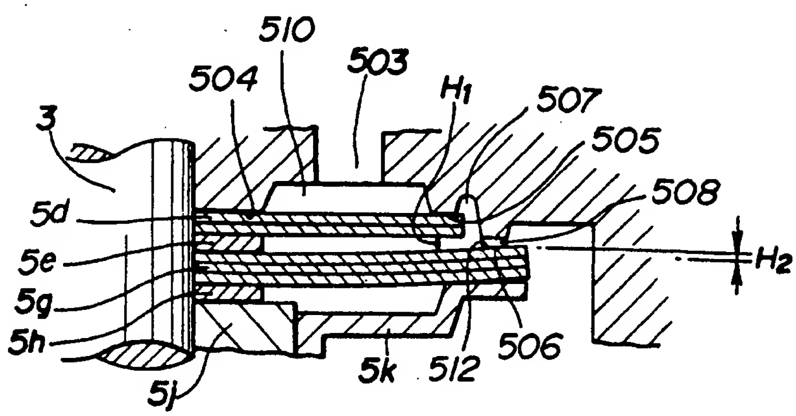

510
503
504
H_1
507
3
505
508
5d
5e
H_2
5g
5h
506
5k
512
5j

FIG. 7

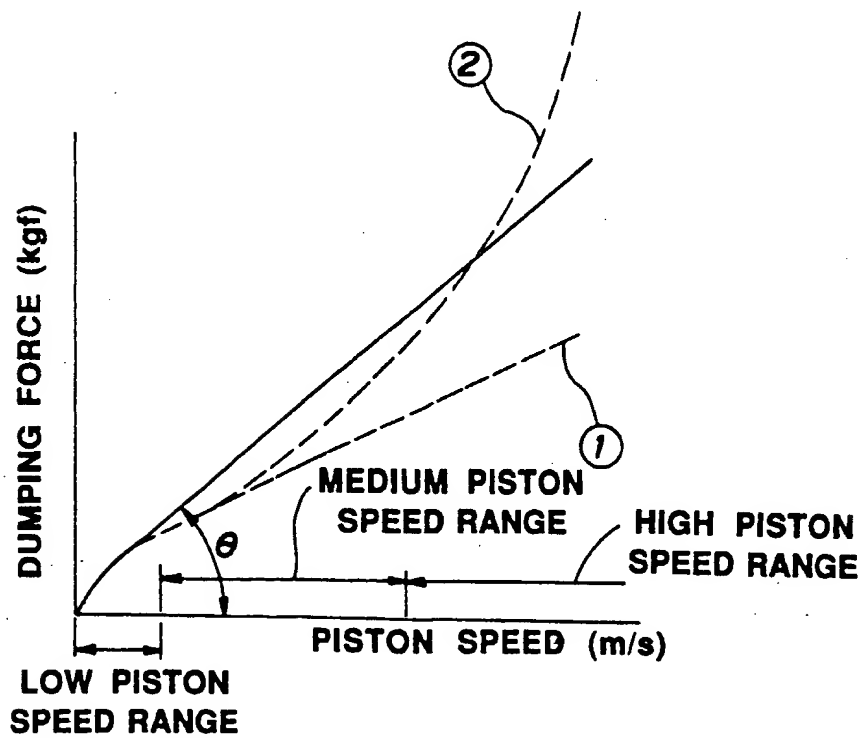

2
DUMPING FORCE (kgf)
1
MEDIUM PISTON SPEED RANGE
HIGH PISTON SPEED RANGE
θ
PISTON SPEED (m/s)
LOW PISTON SPEED RANGE

FIG. 8

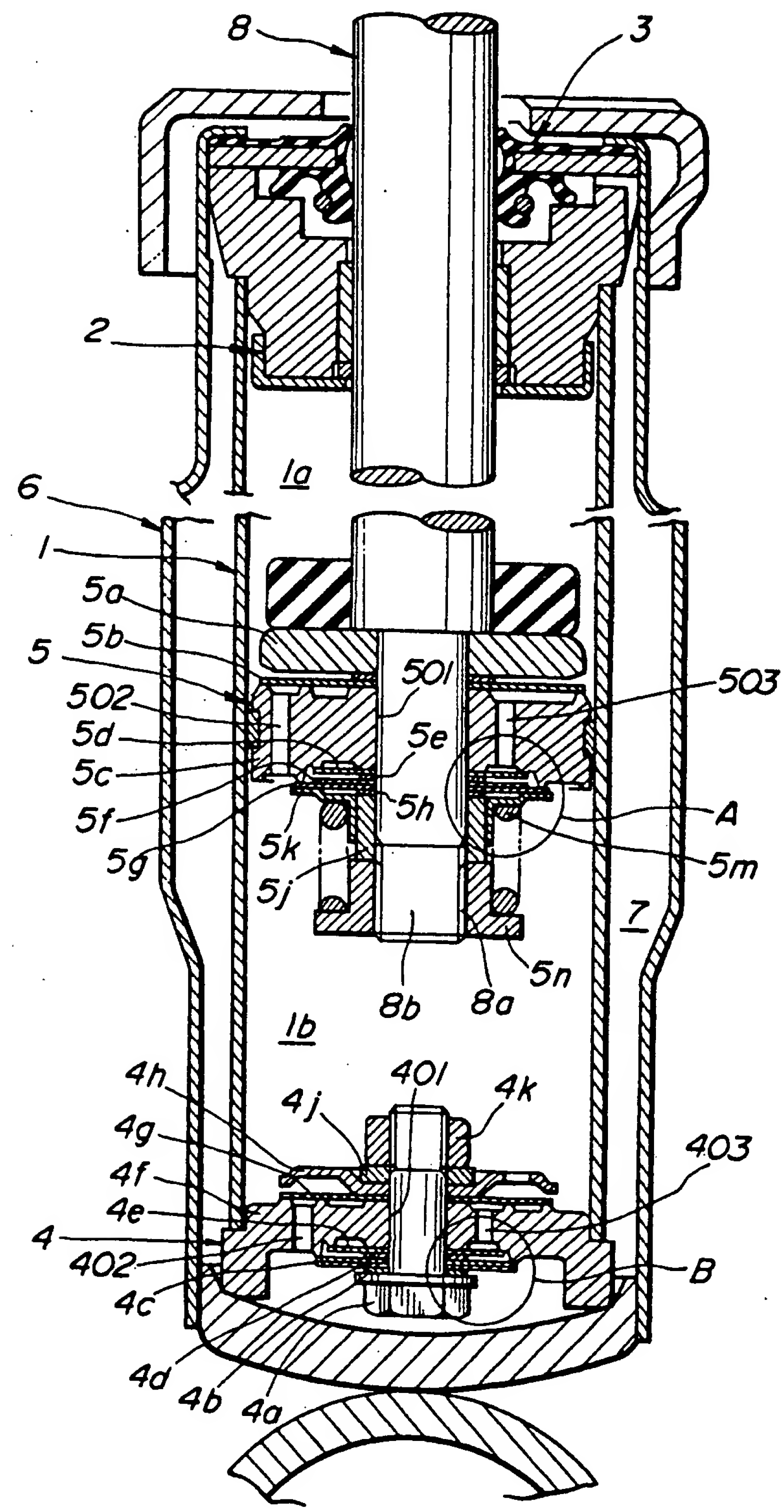

8
3
2
1a
6
1
5a
5b
5
502
5d
5c
5f
5g
501
503
5e
5h
A
5k
5j
5m
7
5n
8b
8a
1b
401
4h
4j
4k
4g
403
4f
4e
4
402
4c
B
4d
4b
4a

FIG. 9

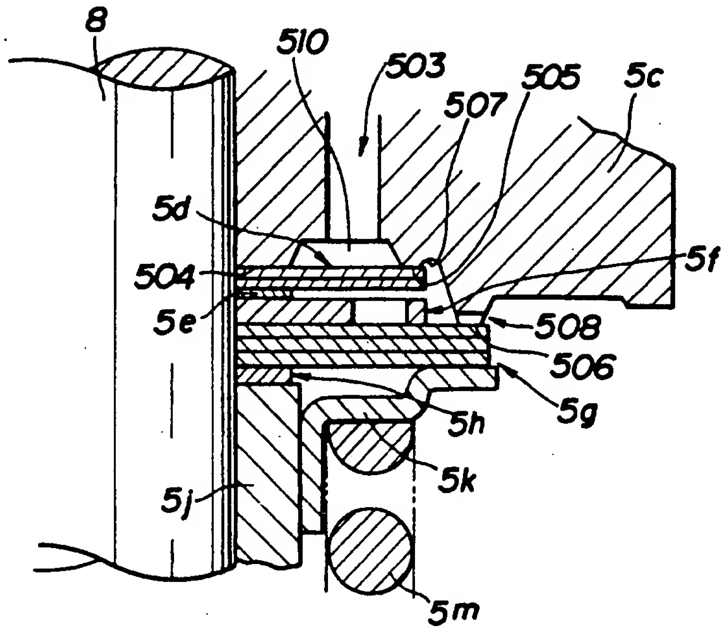

8
510
503
507
505
5c
5d
504
5f
5e
508
506
5g
5h
5k
5j
5m

FIG. 10

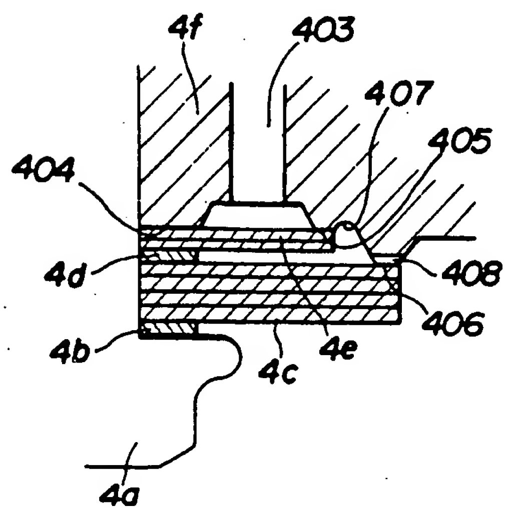
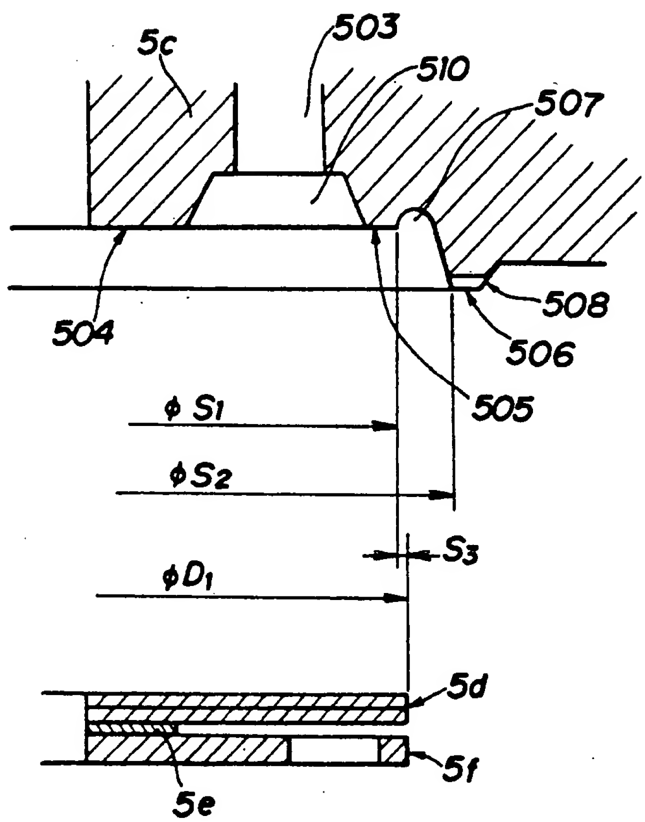

4f
403
407
405
404
4d
408
406
4b
4e
4c
4a

FIG. 11



HYDRAULIC SHOCK ABSORBER WITH PRE-LOADED VALVE FOR LINEAR VARIATION CHARACTERISTICS OF DAMPING FORCE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates generally to a hydraulic shock absorber, suitable for use in an automotive suspension system. More specifically, the invention relates to a shock absorber having improved piston stroke speed dependent damping characteristics.

2. Description of the Background Art

In general, a hydraulic shock absorber generates a damping force determined by pressure difference across a flow restriction valve structure. As will be appreciated, the pressure difference is variable depending upon the magnitude of the flow restriction at the flow restriction valve structure and the working fluid flow rate. Working fluid flow rate is determined by both magnitude and speed of piston stroke.

When the shock absorber employing a constant orifice is used, the damping force varies at a rate substantially proportional to the square of the piston stroke speed. Therefore, the damping force tends to become insufficient at a relatively low piston stroke speed range so as not to generate a sufficient damping force for successfully damping relative displacement of a vehicular body and a road wheel.

In order to improve this, a two stage disc valve strategy has been proposed for generating a damping force for a relatively low speed piston stroke by a first stage valve and for a higher speed piston stroke by a second stage valve. Such a two stage disc valve strategy has been proposed in German Patent 833 574, for example. The proposed shock absorber has the first stage and second stage disc valves arranged in a tandem fashion. The first stage disc valve is principally active for generating a damping force at a relatively low piston stroke speed range. On the other hand, the second stage disc valve is principally active for generating a damping force at a higher piston stroke speed range. Therefore, combining the first and second stage disc valves, improved piston stroke speed dependent damping characteristics can be obtained. Namely, in the aforementioned German Patent, the damping force varies at a rate substantially proportional to a two-thirds ($\frac{2}{3}$) power of the piston stroke speed.

On the other hand, in view of the ease of tuning of an automotive suspension system for achieving both vehicular riding comfort and driving stability, it is desirable to provide a shock absorber having damping characteristics linearly proportional to the piston stroke speed. In view of this requirement, the conventionally proposed shock absorbers are not satisfactory.

SUMMARY OF THE INVENTION

Therefore, it is an object of the present invention to provide a shock absorber which can provide substantially linear damping characteristics in relation to piston stroke speed.

Another object of the invention is to provide a shock absorber having a pre-loaded valve member for providing a higher relief point for providing linear variation characteristics of damping force.

In order to accomplish the aforementioned and other objects, a hydraulic shock absorber, according to the present invention, employs first and second stage disc valves arranged in a tandem fashion. The first stage disc valve is provided for acting on a relatively small pressure difference for enhanced damping characteristics at a relatively low piston stroke speed range. On the other hand, the second stage disc valve is provided for acting on a greater pressure difference for generating a damping force at a higher piston stroke speed range. The second stage disc valve is initially pre-loaded at a predetermined magnitude of load so as to set a pressure relief point of the second stage disc valve at a desired piston stroke speed for better damping characteristics of the shock absorber.

According to one aspect of the invention, a hydraulic shock absorber comprises:

a hollow cylinder filled with a working fluid;

a piston thrustingly disposed within the interior space of the cylinder for defining first and second fluid chambers;

a fluid communication means for establishing fluid communication between the first and second chambers;

a flow restrictive first valve means associated with the fluid communication means for generating a first damping force in response to piston stroke in one direction, the first valve means generating the first damping force according to first variation characteristics when the piston stroke speed is lower than a first criterion and according to second variation characteristics when the piston stroke speed becomes in excess of the first criterion;

a flow restrictive second valve means associated with the fluid communication means and arranged in series with the first valve means, for generating a second damping force in response to piston stroke in the one direction, the second valve means generating the second damping force according to third variation characteristics when the piston stroke is lower than a second criterion and according to fourth variation characteristics when the piston stroke speed becomes in excess of the second criterion; and

means for pre-loading the second valve means for adjusting the second criterion for setting a transition point between the third and fourth variation characteristics.

The first criterion may be set at a lower piston stroke speed than the second criterion. The first variation characteristics may have a greater gradient than that of the second variation characteristics in the piston stroke speed range lower than the first criterion, and the third variation characteristics have a smaller gradient than that of the fourth varying characteristics in the piston speed range lower than the second criterion.

In the preferred construction, the pre-loading means comprises a seat surface offset from the orientation perpendicular to an axis of the shock absorber for forcingly bending the second valve means for exerting a pre-load. Alternatively, the pre-loading means comprises a seat surface offset from the orientation perpendicular to an axis of the shock absorber and the second valve means is provided with a spring force toward the seat surface for self-inducing a pre-load. In the former case, the pre-loading means causes deformation of the second valve means in a direction away from the first valve means for exerting a pre-load. In the latter case, the pre-loading means causes deformation of the second valve for pre-loading, and the second valve means as seated on the seat surface serves as a means for restricting deformation of the first valve means.

The second valve means may be provided with a greater external diameter than the diameter of an outer circumferential edge of a seat surface on which the second valve means is seated, while the piston stroke speed is lower than the second criterion.

In the preferred construction, the first valve means comprises a first window opening defined on the piston and communicated with the fluid path, the window opening being surrounded by a first land having a first surface, and a first resilient valve means resiliently biased toward the surface for normally establishing sealing contact with the first surface and responsive to fluid flow in a first flow direction generated by the piston stroke in the one stroke direction for forming a first flow restrictive path for fluid communication from the first window opening and one of the first and second fluid chambers for generating the first damping force, and a second window opening formed on the piston in fluid communication with the first window opening, the second window opening being defined by a second land with a second surface, and a second resilient valve means resiliently biased toward the second surface for normally establishing sealing contact with the second surface and responsive to fluid flow in a first flow direction generated by the piston stroke in the one stroke direction for forming a second flow restrictive path for fluid communication between the first and second window openings for generating the second damping force.

Preferably, the shock absorber further comprises third and fourth valve means provided for generating a damping force in response to a fluid flow in a second direction opposite to the first direction, the third and fourth valve means being arranged in series and being so designed as to establish essentially linear variation characteristics of damping force depending upon piston stroke speed. In such case, the third valve means may be responsive to piston stroke for generating a third damping force variable according to first variation characteristics in relation to variation of the piston stroke speed in a piston stroke speed range lower than a third criterion and according to second variation characteristics when the piston stroke speed is in excess of the third criterion, and the fourth valve means may be responsive to the piston stroke for generating a fourth damping force variation according to third variation characteristics in relation to variation of the piston stroke speed when the piston stroke speed is lower than a fourth criterion and according to fourth variation characteristics when the piston stroke speed is in excess of the fourth criterion, wherein the third valve means comprises a third window opening defined on the piston and communicated with the fluid path, the window opening being surrounded by a third land having a third surface, and a third resilient valve means resiliently biased toward the surface for normally establishing sealing contact with the third surface and responsive to fluid flow in a second flow direction generated by the piston stroke in the other stroke direction. for forming a third flow restrictive path for fluid communication from the third window opening and one of the first and second fluid chambers for generating the third damping force, and a second window opening formed on the piston in fluid communication with the third window opening, the fourth window opening being defined by a fourth land with a fourth surface, and a fourth resilient valve means resiliently biased toward the fourth surface for normally establishing sealing contact with the fourth surface and responsive to fluid flow in a second flow direction generated by the piston stroke in the other stroke direction for forming a fourth flow restrictive path for fluid communication between the first and fourth window openings for generating the fourth damping force.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be understood more fully from the detailed description given herebelow and from the accompanying drawings of the preferred embodiment of the invention, which, however, should not be taken to limit the invention to the specific embodiment but are for explanation and understanding only.

In the drawings:

FIG. 1 is a section of the preferred embodiment of a shock absorber according to the present invention;

FIG. 2 is an enlarged section of the major part of a piston employed in the preferred embodiment of the shock absorber of FIG. 1, which is the detail in the encircled portion A in FIG. 1;

FIG. 3 is an enlarged section of the major part of a bottom valve employed in the preferred embodiment of the shock absorber of FIG. 1, which illustrates a detailed construction of the encircled portion B in FIG. 1;

FIGS. 4(A), 4(B) and 4(C) are graphs showing variations of damping force relative to piston stroke speed, in which FIG. 4(A) shows damping characteristics of a first stage disc valve relative to the piston stroke speed, FIG. 4(B) shows damping characteristics of a second stage disc valve relative to the piston stroke speed, and FIG. 4(C) shows damping characteristics of flow restriction orifice;

FIG. 5 is a graph variation of damping force generated by the shock absorber in relation to piston stroke speed;

FIG. 6 is an enlarged section of a piston valve assembly in the modified embodiment of shock absorber according to the present invention;

FIG. 7 is a graph showing damping characteristics of the modified embodiment of the shock absorber in comparison with damping characteristics of the conventional shock absorbers;

FIG. 8 is a section of another embodiment of a shock absorber according to the present invention;

FIG. 9 is an enlarged section of the major part of a piston employed in another embodiment of the shock absorber of FIG. 8, which is the detail in the encircled portion A in FIG. 8;

FIG. 10 is a further enlarged section of the major part of a piston valve assembly of FIG. 9, in which is shown a dimensional relationship of the components in the piston valve assembly; and

FIG. 11 is an enlarged section of the major part of a bottom valve employed in the another embodiment of the shock absorber of FIG. 8, which illustrates detailed construction of the encircled portion B in FIG. 8.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings, particularly to FIG. 1, the preferred embodiment of a hydraulic shock absorber, according to the present invention, employs a double action structure including inner and outer cylinders 1 and 6. The top end of the inner cylinder 1 is closed by a guide member 2 and a seal member 3. On the other hand, a bottom fitting assembly 4. Therefore, the inner cylinder 1 defines an enclosed space filled with a working fluid. A piston assembly 5 is disposed within the enclosed space of the inner cylinder 1 for thrusting

movement therein and dividing the enclosed space into upper and low fluid chambers **1**a and **1**b. On the other hand, an annular reservoir chamber **7** is filled with a working fluid and working gas.

The piston assembly **5** is mounted on the lower end of a piston rod **8** for thrusting movement therewith. The piston assembly **5** comprises a retainer **5**a, a check plate **5**b, a piston body **5**c, a first stage disc valve **5**d, a washer **5**e, a stopper plate **5**f, a second stage disc valve **5**g, a washer **5**h, a collar **5**j, a spring seat **5**k and an assist spring **5**m. These components are gathered at the smaller diameter section **8**b of the piston rod **8** and firmly secured to the lower end by means of a fastening nut **5**n which engages with a threaded portion **8**a of the small diameter section **8**b of the piston rod.

The piston body **5**c defines a through opening **502** oriented in the vicinity of the outer circumference thereof. The through opening **502** may be hereafter referred to as an "outer axial opening". The piston body **5**c also defines a through opening **503** oriented at an orientation close to a center opening **501** which receives the small diameter section **8**b of the piston rod **8**. The opening **503** will be hereafter referred to as an "inner axial opening". The upper end of the outer axial opening **502** is openably closed by means of the check plate **5**b. The check plate **5**b blocks fluid flow from the upper fluid chamber **1**a to the lower fluid chamber **1**b. On the other hand, the check plate **5**b is responsive to the fluid pressure in the lower fluid chamber **1**b for permitting fluid flow through a gap formed by deformation of the check plate from the lower fluid chamber **1**b to the upper fluid chamber **1**a.

On the other hand, as shown in FIG. **2**, the lower end of the inner axial opening **503** is closed by the first and second stage disc valves **5**d and **5**g. The first stage disc valve **5**d is normally seated on inner and outer seat surfaces **504** and **505**. A cross-sectionally essentially semi-circular groove **507** is formed adjacent the outer side seat surface **505**. On the other hand, the second stage disc valve **5**g is seated on annular seat surface **506** which is formed along the outer circumference of the piston body **5**c. As can be seen from FIG. **2**, the first stage disc valve **5**d opposes the stopper plate **5**f via the washer **5**e. The circumferential edge of the washer **5**e defines a support for deformation of the first stage disc valve **5**d. The magnitude deformation of the first disc valve **5**d is limited by the stopper plate **5**f so that the maximum deformation magnitude corresponds to the thickness of the washer **5**e. Once the circumferential edge of the first disc valve **5**d comes into contact with the stopper plate, the intermediate portion of the first disc valve is gradually deformed with a progressively increased reaction force.

It should be appreciated that, in the shown embodiment, the first disc valve is provided with a relatively low spring constant so that it may react to a substantially small pressure difference between the upper and lower fluid chambers **1**a and **1**b. Therefore, even at a very low piston speed, the first disc valve **5**d is deformed for permitting a corresponding flow rate of fluid flow for generating a damping force.

As seen from FIG. **2**, the contact point **512** between the second stage disc valve **5**g and the annular seat surface **506** is oriented at a downwardly offset position at a magnitude H_1 from the orientation of the lower surface of the stopper plate. On the other hand, one or more constant orifices **508** are formed between the second disc valve **5**g and the seat surface **507** to permit minimum fluid flow. The constant orifice **508** may not be effective at an initial stage of the piston stroke until the first disc valve **5**d is deformed at a given magnitude to establish a given fluid flow.

The second disc valve **5**g is provided with a greater spring constant so as to provide greater resistance in deformation. The spring coefficient of the second disc valve **5**g is so determined as to achieve a linear variation of the damping force depending upon the piston stroke magnitude and piston stroke speed.

The bottom fitting is provided with a bottom valve assembly **4**. The bottom valve assembly **4** comprises outer and inner axial openings **402** and **403** defined through a body **4**f of the fitting. The valve assembly also comprises a washer **4**b, a second stage disc valve **4**c, a washer **4**d, a first stage disc valve **4**e, a check plate **4**g, a check spring **4**h and a collar. These components are gathered and secured onto the fitting body **4**f by means of fastening bolt **4**a, for which fastening nut **4**k is engaged. The upper end of the outer axial opening **402** is operably closed by the check plate **4**g by seating onto seat surfaces defined on the upper surface of the fitting. Therefore, the fluid flow from the lower fluid chamber **1**b to the reservoir chamber **7** is blocked and the fluid flow in the opposite direction is permitted.

On the other hand, as shown in FIG. **3**, the first disc valve **4**e openably closes the lower end of the inner axial opening **403** by seating onto the seat surfaces **404** and **405** respectively defined on the center boss section and an annular land extending circumferentially at the radially outer side of the inner axial opening **403**. An essentially semi-circular groove **407** is formed immediately outside of the seat surface **405** and extends therealong. The second disc valve **4**c is seated onto a seat surface **406** at the circumferential edge portion.

As seen from FIG. **3**, the contact point **412** between the second stage disc valve **4**c and the annular seat surface **406** is oriented at a downwardly offset position at a magnitude H_2 from the orientation of the lower surface of the stopper plate. One or more constant orifices **408** is formed through the seat surface **406** so as to provide a constant fluid flow at a minimum flow rate.

As can be seen from FIGS. **2** and **3**, the bottom valve assembly operates substantially in the same manner to that valve assembly in the piston.

The operation of the shown embodiment of the shock absorber will be discussed herebelow with respect to respective of rebounding and bounding mode operations.

In the piston bounding mode stroke, the piston assembly **5** moves upwardly relative to the inner cylinder **1** for compressing the volume of the upper fluid chamber **1**a and expanding the volume of the lower fluid chamber **1**b. By variation of the volumes, a fluid pressure difference is generated so that the fluid pressure in the upper fluid chamber **1**a becomes higher than in the lower fluid chamber **1**b. Therefore, fluid flow from the upper fluid chamber **1**a to the lower fluid chamber **1**b is generated. Furthermore, because of lowering of the fluid pressure in the lower fluid chamber **1**b, the fluid pressure in the reservoir chamber **7** becomes higher than that in the lower fluid chamber **1**b for causing fluid flow through the bottom valve assembly. Therefore, working fluid in the upper fluid chamber **1**a and the reservoir chamber **7** flows into the lower fluid chamber **1**b until the pressure balance between the upper and lower fluid chambers and the reservoir chamber is established.

During the piston rebounding stroke, the working fluid in the upper fluid chamber **1***a* flows into the inner axial opening **503**. Against the fluid flow, the first and second disc valves **5***d* and **5***g* are active for providing fluid flow restriction and thus generating damping force. FIGS. **4**(A) and **4**(B) show damping characteristics of the respective individual first and second stage disc valves **5***d* and **5***g*, in relation to the piston stroke speed. As can be seen from FIG. **4**(A), the first stage disc valve **5***d* is normally in a closed position for completely blocking fluid flow from the upper fluid chamber **1***a* to the lower fluid chamber **1***b*. The first stage disc valve **5***d* is responsive to even a relatively small pressure difference to cause deformation for forming a fluid flow orifice between the seat surface **506** to permit a limited amount of fluid flow from the upper fluid chamber to the lower fluid chamber. As a result, damping force is created as shown in FIG. **4**(A). As can be seen, at the initial stage of the piston stroke, the damping force is increased in proportion to the piston stroke speed S in a rate of two-thirds power of the piston speed ($S^{(2/3)}$) (as in the range (1) of FIG. **4**(A). The damping force generated by the first stage disc valve is much greater than that generated by the constant orifice in the prior art. On the other hand, in the low piston stroke speed range, the second stage disc valve **5***g* is held seated on the associated seat surface **506**. Therefore, during the low piston stroke speed range, only the constant orifices **508** are active for generating damping force. Therefore, as shown in the region (2) in FIG. **4**(B), the second stage disc valve **5***g* generates damping force varying at a rate proportion to two power of the piston stroke speed (S^2).

When the circumferential edge of the first disc valve **5***d* comes into contact with the stopper plate **5***f*, the spring constant of the first disc valve becomes greater to vary the variation rate of the damping force to be at a greater rate. In FIG. 4(A), the point where the variation characteristics of the damping force is changed corresponds to the magnitude of the pressure difference at which the circumferential edge of the first disc valve comes into contact with stopper plate **5***f*. When the pressure difference has grown greater than the point above the relief point of the first stage disc valve **5***d*, the flow restriction path formed by the first stage disc valve **5***d* becomes substantially constant. As a result, the variation characteristics of the damping force relative to variation of the piston stroke speed becomes substantially corresponding to that of the constant orifice as shown in the region (3) of FIG. 4(A).

As set forth, the conventional constant orifice provides variation characteristics of the pressure difference proportional to the square of the fluid flow rate Q (Q^2). On the other hand, the variation characteristics of the pressure difference in the first disc valve of the invention is proportional to the two-thirds power of the fluid flow rate ($Q^{(2/3)}$). As seen, in the shown embodiment, a relatively large damping force is generated at very initial stage of the piston stroke.

On the other hand, FIG. **4**(B) shows the variation characteristics of the damping force versus the piston stroke speed at the second stage disc valve **5***g*. As set forth above, the second stage disc valve **5***g* is held at a closed position in the low piston stroke range. At this condition, the working fluid flows through the constant orifices **508**. In the low piston stroke speed range, since the only constant orifice **508** is effective for generating the damping force, the variation characteristics of the damping force in the low piston stroke range becomes substantially proportional to square of the piston stroke speed S (S^2) as shown in region (2) of FIG. **4**(B).

When the piston stroke speed is increased to increase pressure difference, a greater force is exerted on the second stage disc valve **5***g* for causing the latter to deform for increasing the fluid flow path area. The fluid pressure difference at which the second stage disc valve starts to deform is referred as a relief point p. As is clear from FIG. **4**(B), the damping force increasing rate at the second stage disc valve **5***g* is thus small in the low piston stroke speed range. After reaching the turning point p where the second stage disc valve **5***g* starts to open, the variation characteristics becomes substantially proportional to two-thirds power of the piston speed as shown in the region (**4**) of FIG. **4**(B).

In addition to the above, the inner axial path **503** serves as a constant orifice for generating an additional damping force. As can be seen from FIG. **4**(C), since the path area of the inner axial path **503** is held constant, the variation characteristics to be generated by this inner axial path **503** is substantially proportional to the square of the piston stroke speed.

Therefore, by a combination of the first stage and second stage disc valves **5***d* and **5***g*, and the inner axial opening **503**, essentially linear variation characteristics as shown in FIG. **5** can be provided. Such linear characteristics of the variation of the damping force provided by the preferred embodiment of the shock absorber is effective for obtaining better vehicular body attitude stabilization capacity when the shock absorber is applied as a component of the automotive suspension system, with satisfactorily high response. Particularly, the invention is particularly effective in damping relatively low speed piston stroke. Furthermore, according to the shown embodiment, since the variation characteristics of the damping force is essentially linear in the shown embodiment, high vehicular driving stability can be obtained.

Furthermore, in order to enhance the damping characteristics in the piston stroke speed, the relief point p of the second stage disc valve **5***g* is to be risen. In order to set the relief point p at a higher point (higher pressure difference), it is preferable to provide higher initial resiliency for the second stage disc valve without changing a spring coefficient thereof. For this, the shown embodiment provides a predetermined magnitude of pre-load for providing an initial deformation of the second stage disc valve **5***g* by shifting or offsetting the seat point between the seat surface **506** and the second stage disc valve. By providing a pre-load, a resilient force of the second stage disc valve at the initial position resisting against the deformation force exerted thereonto by pressure difference between the upper and lower fluid chamber, is increased. As a result, the relief point p becomes higher as shown by broken line and point p', as shown in the region (6) of FIG. **4**(B). Therefore, by adjusting the offset magnitude H_1, the relief point p can be set at a desired point. This can be compared with the characteristics obtained from the conventional construction as shown by the one-dotted line b of FIG. **4**(B).

On the other hand, in the piston bounding stroke, the piston strokes while compressing the lower fluid chamber **1***b* generate a fluid pressure difference between the upper and lower fluid chambers and between the lower fluid chamber **1***b* and the fluid reservoir chamber **7**. As a result, fluid flow toward the upper fluid chamber **1***a* and toward the fluid reservoir chamber **7** from the

lower fluid chamber **1b** is generated. Then, the first and second stage disc valves **4f** and **4c** acting as third and fourth valve means for the assembly, become effective for generating damping a force varying according to generally linear characteristics as set forth with respect to the valve assembly of the piston.

During this piston bounding stroke, the piston ring **11** and the seal ring **12** are effective for assuring a leak tight seal for avoiding lowering of the damping force at the initial stage of the piston stroke.

Similarly to the foregoing piston valve assembly, by adjusting an offset magnitude H_2, the relief point of the second stage disc valve **4c** can be adjusted for obtaining desired damping characteristics. Therefore, desired damping characteristics can be obtained for damping bounding piston stroke.

FIG. **6** shows a modification of the foregoing first embodiment of the shock absorber according to the present invention. As can be seen from FIG. **6**, the shown embodiment is constructed by neglecting the stopper plate for restricting magnitude of deformation of the first stage disc valve **5d**. Therefore, in this embodiment, the second stage disc valve **2g** serves as a stopper for restricting deformation magnitude of the first stage disc valve.

According to the shown embodiment, in order to adjust the deformation stroke H_1' of the first stage disc valve **5d**, the seating point **512** between the seat surface **506** and the second stage disc valve **5g** is offset upwardly in a magnitude H_2'. By this construction, in the medium piston stroke range in which the first stage disc valve **5d** is held in a fully open position and the second stage disc valve **5g** is still held in a closed position, a greater rate of variation of damping force in relation to the piston stroke speed can be obtained. On the other hand, after starting deformation of the second stage valve **5g**, the first stage disc valve **5d** is again permitted to deform for lowering the increasing rate of the damping force. At the same time, since the second stage disc valve is deformed, the damping characteristics at the second stage disc valve become proportional to the two-thirds ($\frac{2}{3}$) power of the piston stroke speed. As a result, in the high piston stroke speed range, variation rate of the damping force relative to the piston stroke speed becomes smaller.

Therefore, the damping characteristics as shown by solid line in FIG. **7** can be obtained. In FIG. **7**, the damping characteristics obtained by the shown embodiment is compared with the characteristics as shown by broken lines **(1)** and **(2)**. The broken line **(1)** shows damping characteristics of in the conventional disc valve which is not limited to the deformation magnitude. On the other hand, the broken line **(2)** shows an example of damping characteristics obtained by adjusting the axial opening flow path area. In such case, though the greater variation rate of the damping force can be obtained in the medium piston speed range, it becomes possible to provide a smaller damping force variation rate. Therefore, improved piston stroke speed dependent damping characteristics can be obtained in the shock absorber.

FIGS. **8** through **11** shows another embodiment of the shock absorber according to the present invention. In this embodiment the first stage disc valve **5d** and the stopper plate **5f** are formed to have an external diameter ϕD_1 which is greater than the external diameter ϕS_1 of the seat surface **505** so that the circumferential edges extend from the outer circumferential edge of the seat surface **505** in a magnitude of S_3. Similarly, the first stage disc valve **4e** of the bottom valve is also provided with greater external diameter than that of the mating seat surface.

With this construction, the extra margin provided for the first stage disc valve may compensate tolerance in formation of the seat surface or the first disc valve and thus assure seating contact.

In addition, by providing semi-circular grooves **507** and **407** for the piston body **5c** and the fitting body **4f**, the stiffness at inner and outer circumferential edges of the seat surface becomes even in sintering process for providing higher cavitation resistance.

While the present invention has been disclosed in terms of the preferred embodiment in order to facilitate a better understanding of the invention, it should be appreciated that the invention can be embodied in various ways without departing from the principle of the invention. Therefore, the invention should be understood to include all possible embodiments and modifications to the shown embodiments which can be embodied without departing from the principle of the invention set out in the appended claims.

What is claimed is:

1. A hydraulic shock absorber comprising:
- a hollow cylinder filled with a working fluid;
- a piston thrustingly disposed within the interior space of said cylinder for defining first and second fluid chambers;
- a fluid communication means for establishing fluid communication between said first and second chambers;
- a flow restrictive first valve means associated with said fluid communication means for generating a first damping force in response to piston stroke in one direction, said first valve means generating said first damping force according to a first variation characteristic when the piston stroke speed is lower than a first criterion and according to a second variation characteristics when the piston stroke speed becomes in excess of said first criterion;
- a flow restrictive second valve means associated with said fluid communication means and arranged in series with said first valve means, for generating a second damping force in response to piston stroke in said one direction, said second valve means generating said second damping force according to a third variation characteristic when the piston stroke speed is lower than a second criterion and according to a fourth variation characteristic when the piston stroke speed becomes in excess of said second criterion; and
- means for pre-loading said second valve means for adjusting said second criterion for setting a transition point between said third and fourth variation characteristics, said pre-loading means deforming a circumferential edge portion of said second valve means.

2. A shock absorber as set forth in claim **1**, wherein said first criterion is set at a lower piston stroke speed than said second criterion.

3. A shock absorber as set forth in claim **1**, wherein said first variation characteristic has a greater gradient than that of said second variation characteristic in the piston stroke speed range lower than the first criterion, and said third variation characteristic has a smaller gradient than that of said fourth variation characteristic

in the piston speed range lower than the second criterion.

4. A shock absorber as set forth in claim 1, wherein said pre-loading means comprises a seat surface offset from an orientation perpendicular to an axis of the shock absorber for forcingly bending the circumferential edge portion of the second valve means for exerting said pre-loading.

5. A shock absorber as set forth in claim 1, wherein said pre-loading means comprises a seat surface offset from an orientation perpendicular to an axis of the shock absorber for forcingly bending the second valve means for exerting said pre-loading.

6. A shock absorber as set forth in claim 4, wherein said pre-loading means causes deformation of said second valve means in a direction away from said first valve means for exerting said pre-loading.

7. A shock absorber as set forth in claim 5, wherein said pre-loading means causes deformation of said second valve for pre-loading, and said second valve means as seated on said seat surface serving as means for restricting deformation of said first valve means.

8. A shock absorber as set forth in claim 1, wherein said second valve means is provided with a greater external diameter than the diameter of an outer circumferential edge of a seat surface, on which said second valve means is seated while the piston stroke speed is lower than said second criterion.

9. A shock absorber as set forth in claim 1, wherein said first valve means comprises a first window opening defined on said said piston and communicated with said fluid path, said window opening being surrounded by a first land having a first surface, and a first resilient valve means resiliently biased toward said surface for normally establishing sealing contact with said first surface and responsive to fluid flow in a first flow direction generated by the piston stroke in said one stroke direction for forming a first flow restrictive path for fluid communication from said first window opening and one of said first and second fluid chambers for generating said first damping force, and a second window opening formed on said piston in fluid communication with said first window opening, said second window opening being defined by a second land with a second surface, and a second resilient valve means resiliently biased toward said second surface for normally establishing sealing contact with said second surface and responsive to fluid flow in a first flow direction generated by the piston stroke in said one stroke direction for forming a second flow restrictive path for fluid communication between said first and second window openings for generating said second damping force.

10. A shock absorber as set forth in claim 1, which further comprises third and fourth valve means provided for generating damping force in response to a fluid flow in a second direction opposite to said first direction, said third and fourth valve means being arranged in series and being so designed as to establish essentially linear variation characteristics of damping force depending upon piston stroke speed.

11. A shock absorber as set forth in claim 10, wherein said said third valve means is responsive to piston stroke for generating a third damping force variable according to a first variation characteristic in relation to variation of the piston stroke speed in a piston stroke speed range lower than a third criterion and according to a second variation characteristic when the piston stroke speed is in excess of said third criterion, and said fourth valve means is responsive to the piston stroke for generating fourth damping force variation according to a third variation characteristics in relation to variation of the piston stroke speed when said piston stroke speed is lower than a fourth criterion and according to a fourth variation characteristics when said piston stroke speed is in excess of said fourth criterion, third valve means comprises a third window opening defined on said said piston and communicated with said fluid path, said window opening being surrounded by a third land having a third surface, and a third resilient valve means resiliently biased toward said surface for normally establishing sealing contact with said third surface and responsive to fluid flow in a second flow direction generated by the piston stroke in the other stroke direction for forming a third flow restrictive path for fluid communication from said third window opening and one of said first and second fluid chambers for generating said third damping force, and a second window opening formed on said piston in fluid communication with said third window opening, said fourth window opening being defined by a fourth land with a fourth surface, and a fourth resilient valve means resiliently biased toward said fourth surface for normally establishing sealing contact with said fourth surface and responsive to fluid flow in a second flow direction generated by the piston stroke in said the other stroke direction for forming a fourth flow restrictive path for fluid communication between said first and fourth window openings for generating said fourth damping force.

12. A shock absorber as set forth in claim 1, wherein said pre-loading means includes at least a flow path therethrough for allowing passage of said working fluid.

* * * * *

US005529154A

United States Patent [19]

Tanaka

[11] Patent Number: 5,529,154

[45] Date of Patent: Jun. 25, 1996

[54] VALVE STRUCTURE FOR DAMPER

[75] Inventor: **Kazuhiko Tanaka**, Tochigi, Japan

[73] Assignee: **Showa Corporation**, Saitama, Japan

[21] Appl. No.: **349,814**

[22] Filed: **Dec. 6, 1994**

[30] **Foreign Application Priority Data**

Dec. 6, 1993 [JP] Japan 5-305598
Sep. 14, 1994 [JP] Japan 6-220606

[51] **Int. Cl.**^{6} **F16F 9/348**

[52] **U.S. Cl.** **188/322.15**; 188/280

[58] **Field of Search** 188/280, 282, 188/317, 322.15

[56] **References Cited**

U.S. PATENT DOCUMENTS

4,895,229	1/1990	Kato	188/322.15 X
4,964,493	10/1990	Yamaura et al.	188/322.15
5,085,300	2/1992	Kato et al.	188/317 X
5,115,892	5/1992	Yamaoka et al.	188/322.18 X

FOREIGN PATENT DOCUMENTS

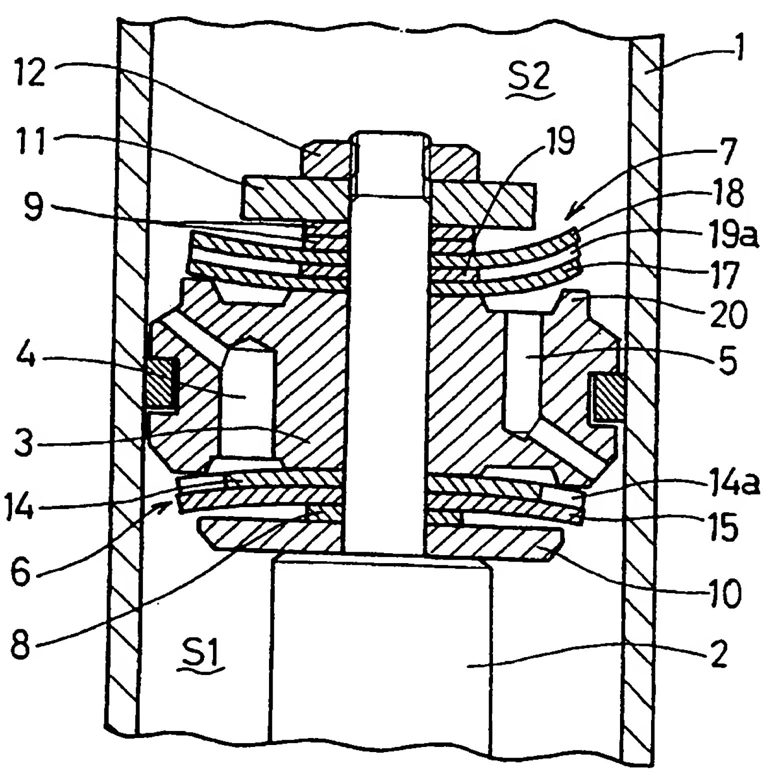
60-99341	7/1985	Japan .	
4185929	7/1992	Japan	188/322.15
2225409	5/1990	United Kingdom	188/322.15

Primary Examiner—Josie Ballato
Attorney, Agent, or Firm—Merchant, Gould, Smith, Edell, Welter & Schmidt

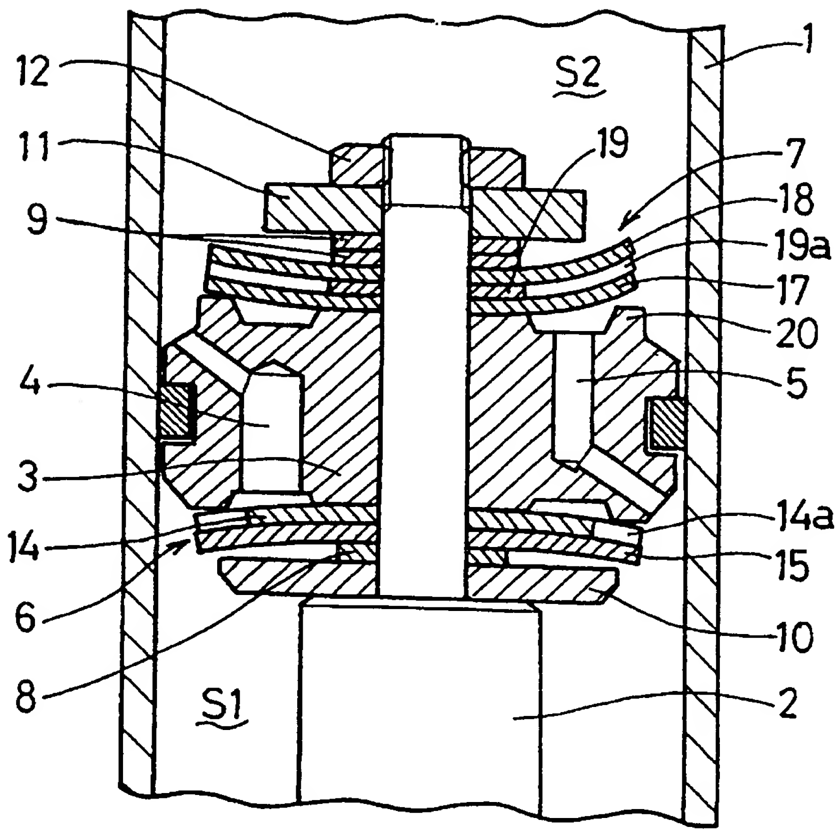
[57] **ABSTRACT**

A valve structure for use in a damper has a cylinder, a piston rod axially movably disposed in the cylinder, a piston or dividing member mounted on an end of the piston rod in the cylinder and dividing an interior space of the cylinder into two oil chambers one on each side of the dividing member, the dividing member having a valve seat, and a circular resilient valve assembly supported at a circumferential edge thereof on the dividing member and having a free end seatable on the valve seat. The circular resilient valve assembly includes a circular auxiliary resilient valve, a circular intermediate resilient sheet, and a circular main resilient valve which have identical inside and outside diameters, the circular auxiliary resilient valve, the circular intermediate resilient sheet, and the circular main resilient valve being stacked in the order named successively from the valve seat. The circular intermediate resilient sheet has at least one recess defined therein and opening at a free end thereof for allowing the circular auxiliary resilient valve to flex. The circular resilient valve assembly can flex depending on a differential pressure between the two oil chambers to produce a damping oil path for applying a damping force against axial movement of the piston rod with respect to the cylinder.

17 Claims, 7 Drawing Sheets



F I G. 1



F I G. 2

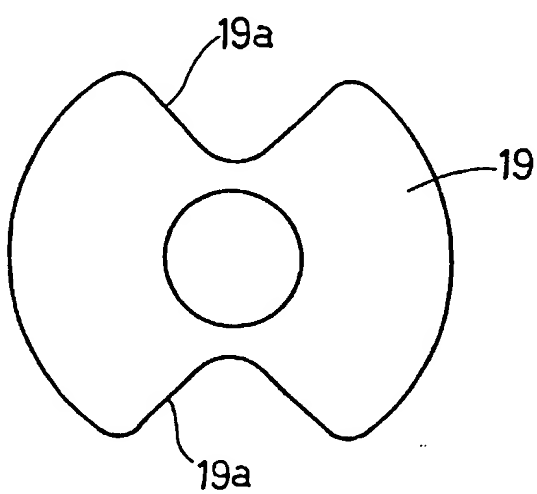


FIG. 3 (a)

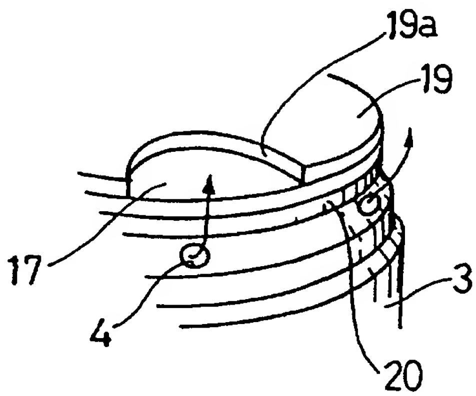


FIG. 3 (b)

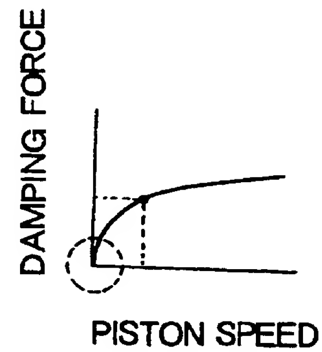


FIG. 4 (a)

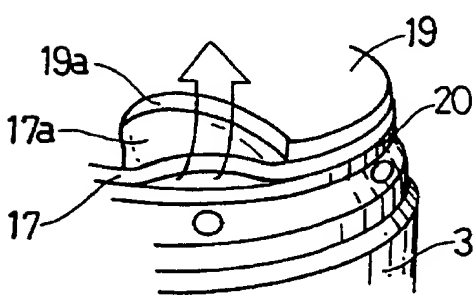


FIG. 4 (b)

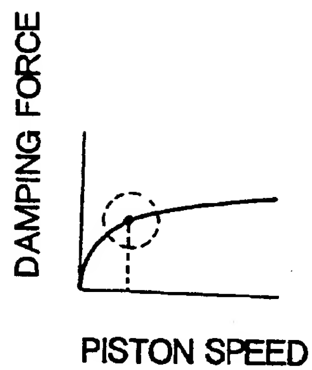


FIG. 5 (a)

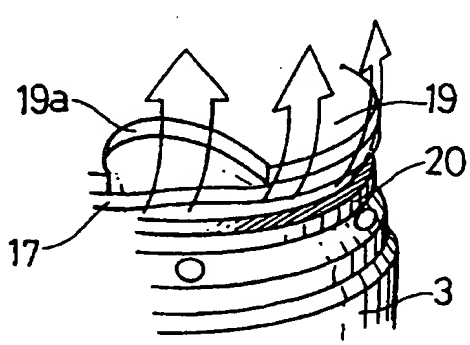


FIG. 5 (b)

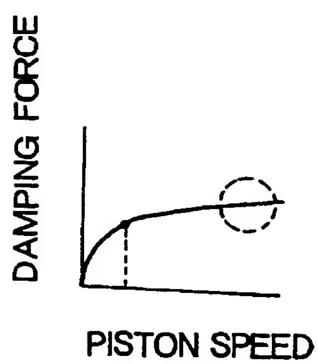


FIG. 6

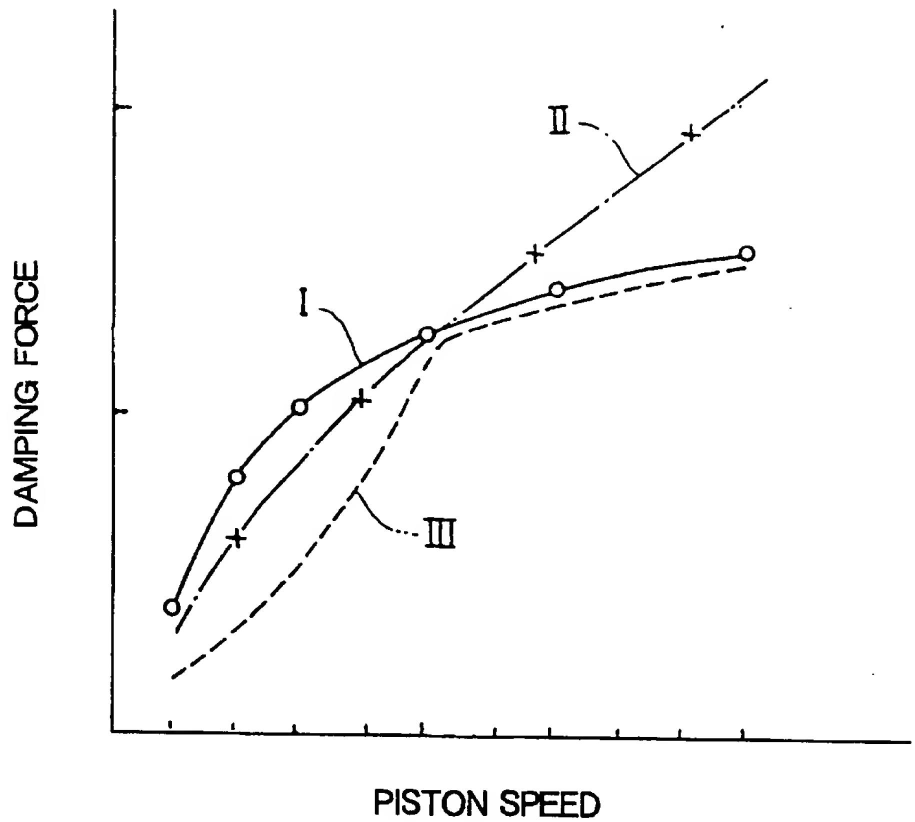


FIG. 7 (a)

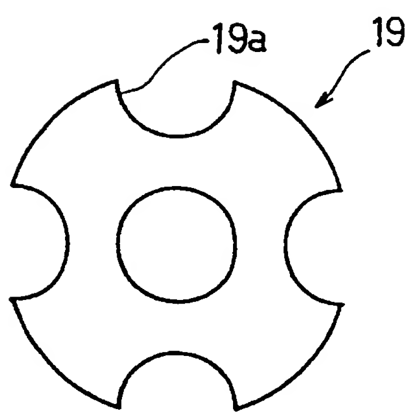


FIG. 7 (b)

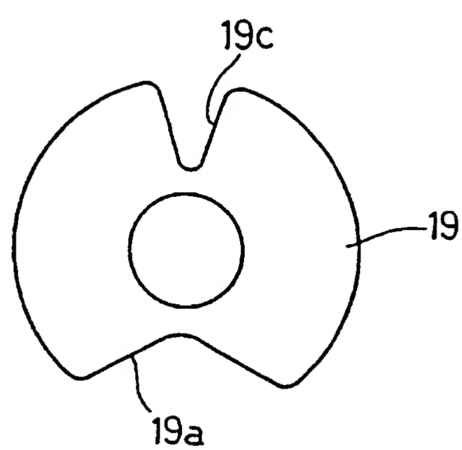


FIG. 8

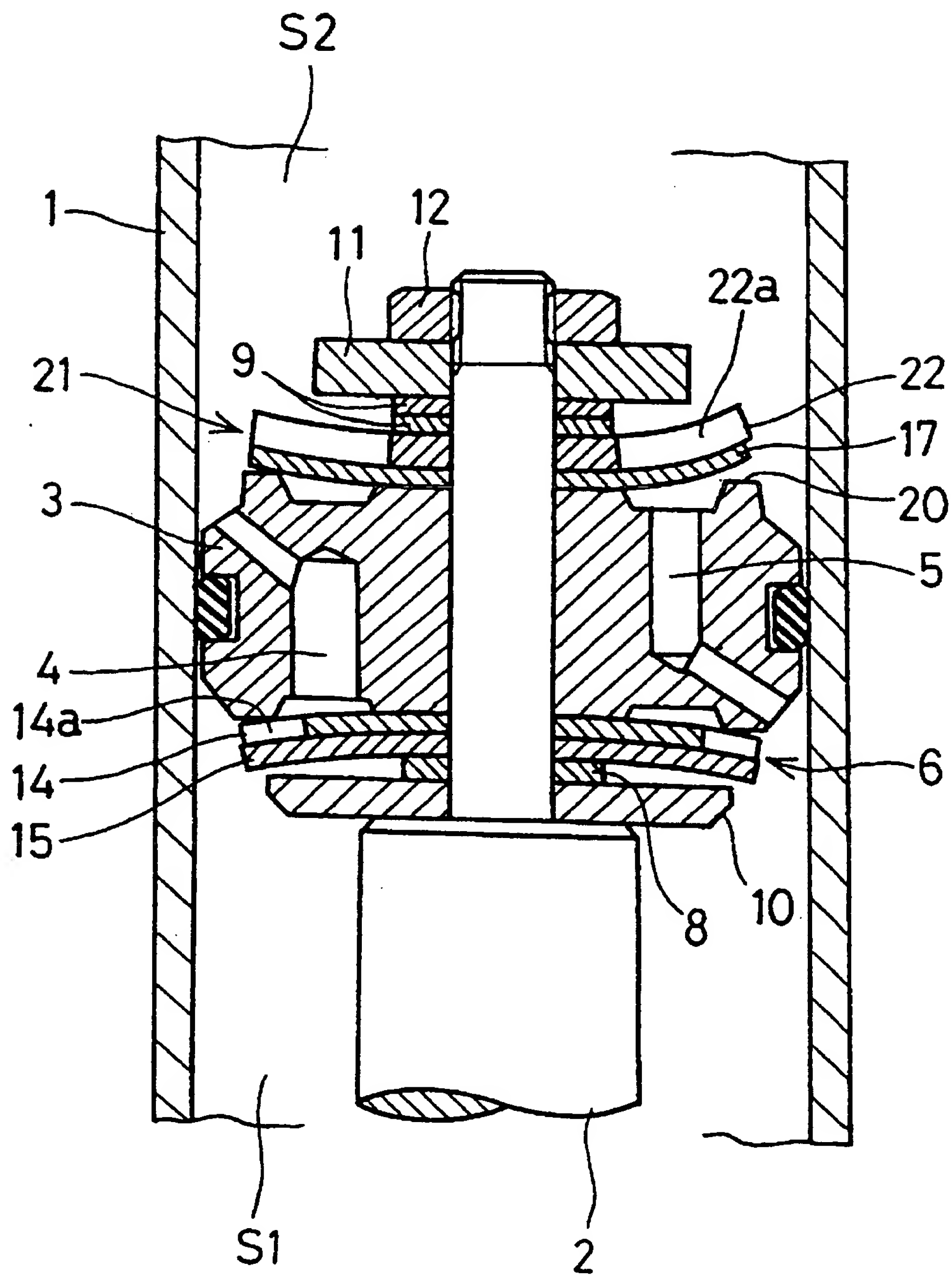


FIG. 9

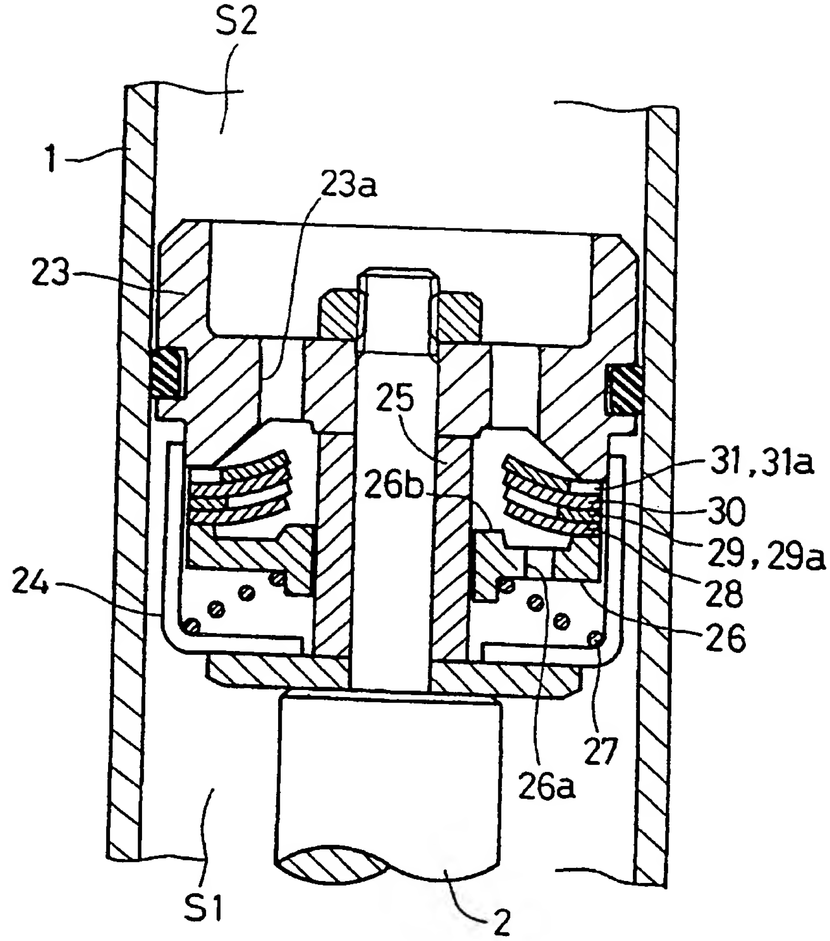


FIG. 10

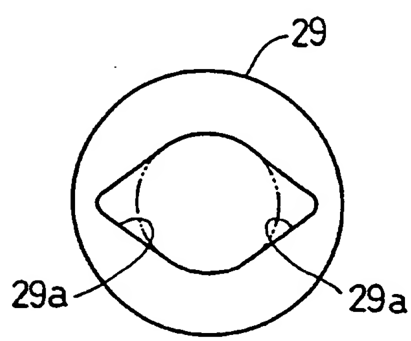


FIG. 11 (a)

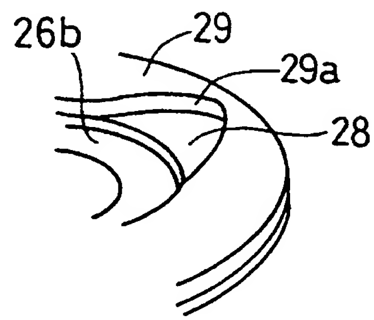


FIG. 11 (b)

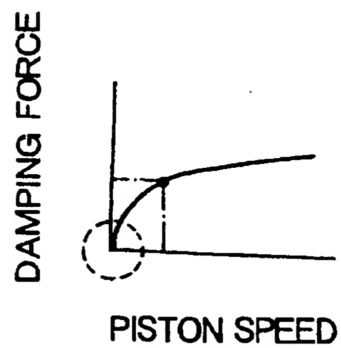


FIG. 12 (a)

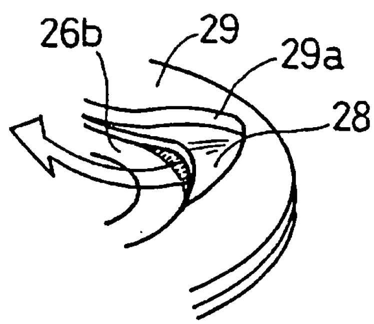


FIG. 12 (b)

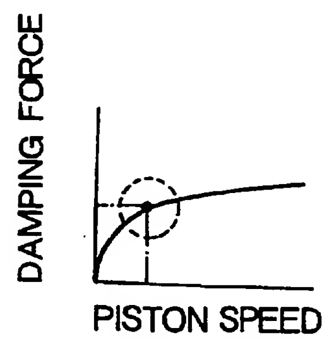


FIG. 13 (a)

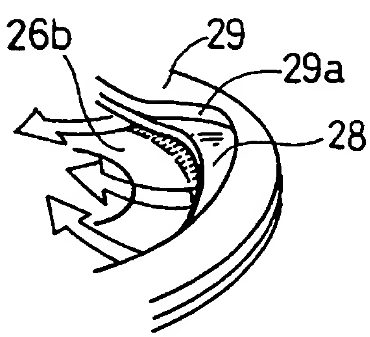


FIG. 13 (b)

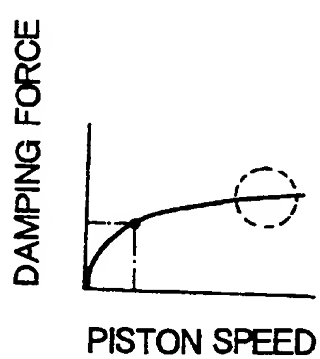


FIG. 14
(PRIOR ART)

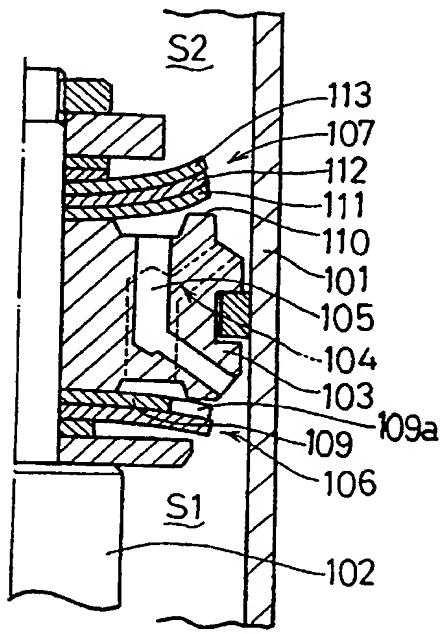


FIG. 15 (a)
(PRIOR ART)

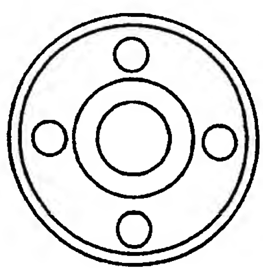


FIG. 15 (b)
(PRIOR ART)

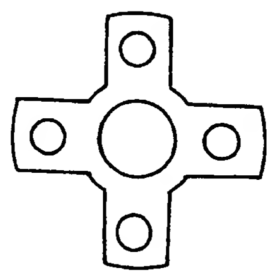
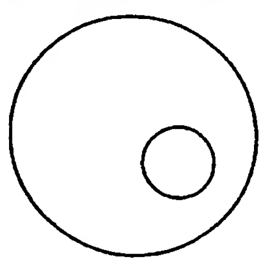


FIG. 15 (c)
(PRIOR ART)



VALVE STRUCTURE FOR DAMPER

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a damper, and more particularly to a valve structure for generating damping forces as a piston of a damper moves back and forth.

2. Description of the Prior Art

FIG. 14 of the accompanying drawings illustrates a conventional hydraulic damper. As shown in FIG. 14, the conventional hydraulic damper has a cylinder **101** and a piston rod **102** extending axially therein. A piston **103** is mounted on a distal end of the piston rod **102** in the cylinder **101** in sliding contact with an inner circumferential surface of the cylinder **101**. The piston **103** divides the interior space of the cylinder **101** into two oil chambers **S1** and **S2**, one on each side of the piston **103**. The piston **103** has a compression oil passage **104** and an expansion oil passage **105** which are defined therein. A compression valve **106** for opening and closing the compression oil passage **104** and an expansion valve **107** for opening and closing the expansion oil passage **105** are disposed one on each side of the piston **103**. The compression valve **106** has a slit **109***a* defined therein. The expansion valve **107** comprises a plurality of plate valves **111, 112, 113** positioned adjacent to a valve seat **110** of the piston **103** remote from the compression valve **106**.

Each of the plate valves **111, 112, 113** is of a circular shape, and the valve seat **110** for bearing these plate valves **111, 112, 113** is of a circular (annular) shape as shown in FIG. **15**(*a*) of the accompanying drawings. The valve structure of the conventional hydraulic damper has such damping force characteristics in an expansion stroke that, as indicated by the curve III in FIG. **6**, when the speed of travel of the piston **103** is low, working oil in the oil chamber S1 flows through the slit **109***a* of the compression valve **109** and the compression oil passage **104** into the oil chamber S2, producing a relatively steep damping force characteristic curve, and when the piston **103** travels at a medium speed, working oil flowing into the expansion oil passage **105** forces the expansion valve **107** to open, producing a relatively gradual damping force characteristic curve because of the circular (annular) valve seat **110**. Therefore, the circular (annular) valve seat **110** fails to provide linear damping force characteristics due to an inflection between the low- and medium-speed ranges.

Another hydraulic damper employs a noncircular (nonannular) valve seat as shown in FIG. **15**(*b*) of the accompanying drawings for opening the valve gradually when the piston speed is low, so that substantially linear damping force characteristics are provided throughout the low- and medium-speed ranges. However, the noncircular valve seat causes the valve to open to a large extent, resulting in relatively steep damping force characteristics in a higher speed range, as shown in FIG. **6**.

Japanese laid-open patent publication No. 60-99341 discloses a hydraulic damper which employs a circular valve seat and an expansion valve similar to the expansion valve **107** shown in FIG. 14. The expansion valve includes an auxiliary plate valve **111**, an intermediate plate valve or sheet **112**, and a main plate valve **113**, the intermediate plate valve **112** having an eccentric shape as shown in FIG. **15**(*c*) of the accompanying drawings. The expansion valve flexes gradually in its fully circumferential region after portions of the auxiliary and main plate valves **111, 113** have started flexing, thereby eliminating any inflection from low- and medium-speed ranges.

While the valve structure disclosed in the above publication provides the same damping force characteristics in the medium-speed range as those of the valve structure with the circular valve seat, it is difficult to grasp the correlation between the configuration of the intermediate plate valve and the generated damping forces.

SUMMARY OF THE INVENTION

It is therefore an object of the present invention to provide a valve structure for a damper which can produce smooth damping force characteristics in an entire speed range of back-and-forth movement of a piston and substantially linear damping force characteristics from low- to medium-speed ranges of the back-and-forth movement of the piston.

According to the present invention, there is provided a valve structure for use in a damper, comprising a cylinder, a piston rod axially movably disposed in the cylinder, a dividing member mounted on an end of the piston rod in the cylinder and dividing an interior space of the cylinder into two oil chambers, one on each side of the dividing member, the dividing member having a valve seat, and a circular resilient valve assembly supported at a circumferential edge thereof on the dividing member and having a free end seatable on the valve seat, the circular resilient valve assembly comprising a circular auxiliary resilient valve, a circular intermediate resilient sheet, and a circular main resilient valve which have identical overall inside and outside diameters, the circular auxiliary resilient valve, the circular intermediate resilient sheet, and the circular main resilient valve being stacked in the order named successively from the valve seat, the circular intermediate resilient sheet having at least one recess defined therein and opening at a free end thereof for allowing the circular auxiliary resilient valve to flex, whereby the circular resilient valve assembly can flex depending on a differential pressure between the two oil chambers to produce a damping oil path for applying a damping force against axial movement of the piston rod with respect to the cylinder.

Alternatively, the circular resilient valve assembly may comprise a circular auxiliary resilient valve and a circular main resilient valve which have identical overall and outside diameters, the circular auxiliary resilient valve and the circular main resilient valve being stacked in the order named successively from the valve seat, the main valve having at least one recess defined therein and opening at a free end thereof for allowing the circular auxiliary resilient valve to flex.

Preferably, the valve seat is of a circular shape, and the recess is of a sectorial shape, a semicircular shape, or a U shape.

The above and further objects, details and advantages of the present invention will become apparent from the following detailed description of preferred embodiments thereof, when read in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a fragmentary cross-sectional view of a damper around its piston, the damper incorporating a valve structure according to a first embodiment of the present invention;

FIG. **2** is a plan view of an intermediate plate valve or sheet in the valve structure shown in FIG. **1**;

FIG. **3**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **1** operates when the piston moves in a low-speed range;

FIG. **3**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **3**(*a*);

FIG. **4**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **1** operates when the piston moves in a medium-speed range;

FIG. **4**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **4**(*a*);

FIG. **5**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **1** operates when the piston moves in a high-speed range;

FIG. **5**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **5**(*a*);

FIG. **6** is a graph showing damping force characteristics of dampers having various valve structures;

FIG. **7**(*a*) is a plan view of a first modification of the intermediate sheet according to the first embodiment;

FIG. **7**(*b*) is a plan view of a second modification of the intermediate sheet according to the first embodiment;

FIG. **8** is a fragmentary cross-sectional view of a damper around its piston, the damper incorporating a valve structure according to a second embodiment of the present invention;

FIG. **9** is a fragmentary cross-sectional view of a damper around its piston, the damper incorporating a valve structure according to a third embodiment of the present invention;

FIG. **10** is a plan view of an intermediate sheet in the valve structure shown in FIG. **9**;

FIG. **11**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **9** operates when the piston moves in a low-speed range;

FIG. **11**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **11**(*a*);

FIG. **12**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **9** operates when the piston moves in a medium-speed range;

FIG. **12**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **12**(*a*);

FIG. **13**(*a*) is a fragmentary perspective view showing the manner in which the valve structure in the damper shown in FIG. **9** operates when the piston moves in a high-speed range;

FIG. **13**(*b*) is a graph showing damping force characteristics of the damper at the time the valve structure operates as shown in FIG. **13**(*a*);

FIG. **14** is a fragmentary cross-sectional view of a damper around its piston, the damper incorporating a conventional valve structure;

FIG. **15**(*a*) is a plan view showing a conventional valve seat shape;

FIG. **15**(*b*) is a plan view showing another conventional valve seat shape; and

FIG. **15**(*c*) is a plan view of a conventional intermediate sheet.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

FIG. **1** shows a damper around its piston, the damper incorporating a valve structure according to a first embodiment of the present invention. As shown in FIG. **1**, the damper includes a cylinder **1** and a piston rod **2** extending wardly therein. A piston (dividing member) **3** is mounted on a distal end of the piston rod **2** in the cylinder **1** in sliding contact with an inner circumferential surface of the cylinder **1**. The piston **3** divides the interior space of the cylinder **1** into two oil chambers **S1** and **S2** one on each side of the piston **3**. The piston **3** has a compression oil passage **4** and an expansion oil passage **5** which are defined therein. A compression valve assembly **6** for opening and closing the compression oil passage **4** and an expansion valve assembly **7** for opening and closing the expansion oil passage **5** are mounted respectively on opposite axial surfaces of the piston **3**. A sheet **8** is disposed behind the compression valve assembly **6**, and a pair of sheets **9** is disposed behind the expansion valve assembly **7**. The piston **3**, the compression valve assembly **6**, the expansion valve assembly **7**, the sheet **8**, and the sheets **9** are axially clamped between retainers **10**, **11** and fastened in place by a nut **12** threaded on the tip end of the piston rod **2**.

The compression valve assembly **6** comprises two inner and outer resilient circular values **14**, **15**, the inner valve **14** having a slit **14***a* defined therein. The expansion valve assembly **7** comprises a circular resilient auxiliary valve **17**, a circular resilient main valve **18**, and a circular resilient intermediate sheet **19** interposed therebetween. The piston **3** has a circular (annular) valve seat **20** on its surface near the expansion valve assembly **7** for abutting contact with the circular auxiliary valve **17**, the valve seat **20** surrounding the expansion oil passage **5**.

The intermediate sheet **19** has an overall outside diameter between recesses which is the same as those of the auxiliary valve **17** and the main valve **18**. As shown in FIG. **2**, the intermediate sheet **19** has a pair of diametrically opposite sectorial recesses **19***a* defined symmetrically therein across the central axis thereof. The sectorial recesses **19***a* serve to allow the auxiliary valve **17** to flex partly under a differential pressure between the oil chambers **S1**, **S2**.

The compression oil passage **4**, the compression valve assembly **6** for opening and closing the compression oil passage **4**, the expansion oil passage **5**, and the expansion valve assembly **7** for opening and closing the expansion oil passage **5** jointly make up a damping force generating means.

When the piston rod **2** moves downwardly as viewed in FIG. **1** in an expansion stroke, working oil in the oil chamber **S1** flows through the slit **14a** into the compression oil passage **4**, and then flows from the compression oil passage **4** into the oil chamber **S2** regardless of the expansion valve assembly **7**, as long as the piston **3** moves in a low-speed range, as shown in FIG. **3**(*a*). Thus, the slit **14***a* serves as an orifice for generating a damping force. Accordingly, as shown in damping force characteristics shown in FIG. **3**(*b*), the damping force gradually increases from a region encircled by a dotted-line circle in FIG. **3**(*b*), depending on the opening of the compression oil passage **4**.

As the piston speed approaches a medium-speed range, working oil flowing into the expansion oil passage **5** forces portions **17***a* of the auxiliary valve **17** to flex axially into the corresponding recesses **19***a* in the intermediate sheet **19**, as shown in FIG. **4**(*a*). Since the portions **17***a* of the auxiliary valve **17** start opening at first, the characteristic curve of the

generated damping force is smooth without infections in a region encircled by a dotted-line circle in FIG. **4**(*b*).

When the piston **3** moves in a high-speed range, the expansion valve assembly **7** opens in its entirety under the pressure of working oil flowing into the expansion oil passage **5**, as shown in FIG. **5**(*a*). At this time, since the valve seat **20** is of a circular shape, the generated damping force gradually increases, without a sharp rise, in a region encircled by a dotted-line circle in FIG. **5**(*b*).

The sectorial recesses **19***a* defined in the intermediate sheet **19** are effective to provide a smooth inflection-free damping force characteristic curve I shown in FIG. **6** even though the expansion valve assembly **7** is circular in shape, with any change in the damping force being gradual in the medium- and high-speed ranges. Since the damping force varies depending on the number, shape, angle, and depth of the recesses **19***a* in the intermediate sheet **19**, it is easy to grasp the correlation between the shape of the intermediate sheet **19** and the damping force.

The recesses **19***a* in the intermediate sheet **19** are not limited to the shape shown in FIG. **2**. As shown in FIG. **7**(*a*), the intermediate sheet **19** may have four semicircular or U-shaped recesses **19***a* angularly spaced at 90°. Alternatively, as shown in FIG. **7**(*b*), the intermediate sheet **19** may have a recess **19***a* defined by two radial sides angular spaced a greater angle from each other and another recess **19***c* defined by two radial sides angular spaced a smaller angle from each other. Further alternatively, the intermediate sheet **19** may have a combination of recesses of different shapes. Instead of the circular intermediate sheet **19**, a noncircular (nonannular) sheet with recesses may be employed. A similar valve arrangement may also be incorporated in the compression valve assembly **6**, or a fixed dividing member such as a bottom valve or the like.

In the first embodiment of the present invention, as described above, the intermediate sheet sandwiched between the main valve and the auxiliary valve is of the same overall outside diameter between recesses as those of the main valve and the auxiliary valve, and has sectorial recesses or recesses of other shapes. The intermediate sheet is effective to provide smooth damping force characteristics in the entire speed range of back-and-forth movement of the piston **3**, substantially linear damping force characteristics from the medium-speed range to the high-speed range, and damping force characteristics determined by the valve seat in the medium- and high-speed ranges. It is also easy to grasp the correlation between the shape of the intermediate sheet **19** and the generated damping force depending on the number, shape, angle, and depth of the recesses **19***a* in the intermediate sheet **19**.

A valve structure according to a second embodiment of the present invention, which is incorporated in a damper, will be described below with reference to FIG. **8**.

As shown in FIG. **8**, the damper on the piston **3** is of substantially the same structure as the damper shown in FIG. **1** except for an expansion valve assembly **21**. The expansion valve assembly **21** comprises an annular resilient auxiliary valve **17** that can be seated on the valve seat **20** of the piston **3**, and a resilient main valve **22** held against a flexing surface of the auxiliary valve **17**. The auxiliary valve **17** and the main valve **22** are concentrically supported on the piston rod **2**. The main valve **22** comprises a resilient sheet having at least one sectorial recess **22***a* defined in a free end thereof as with the intermediate sheet **19** according to the first embodiment. The main valve **22** has a thickness greater than the thickness of the auxiliary valve **17**.

The thicker main valve **22** is partly held in contact with the auxiliary valve **17** for resiliently holding the auxiliary valve **17** against undue flexural displacement. The thicker main valve **22** serves to perform the combined function of the main valve **18** and the intermediate sheet or valve **19** according to the first embodiment. As a result, damping force characteristics in the low- and medium-speed ranges can be set up freely with a relatively simple structure made up of a reduced number of parts.

A valve structure according to a third embodiment of the present invention, which is incorporated in a damper, will be described below with reference to FIGS. **9**, **10**, **11**(*a*), **11**(*b*), **12**(*a*), **12**(*b*), **13**(*a*), and **13**(*b*).

As shown in FIG. **9**, the damper has a piston **23** mounted on a distal end of a piston rod **2** and held in sliding contact with an inner circumferential surface of a cylinder **1**. The piston **23** has a plurality of axial oil passages **23***a* defined therethrough, and a valve guide **24** is concentrically mounted on an axial end of the piston **23** around the piston rod **2**. The valve guide **24** houses therein a collar **25** fitted over the piston rod **2**, a valve holder **26** axially movably disposed between the collar **25** and the valve guide **24**, and a spiral spring **27** acting between the valve holder **26** and the valve guide **24**.

The valve holder **26** has a plurality of axial oil passages **26***a* defined therethrough and a valve seat **26***b* on an inner circumferential edge thereof. An axial stack of a resilient auxiliary valve **28**, a resilient intermediate sheet **29**, a resilient main valve **30**, and a resilient upper valve **31**, which are arranged successively in the order named from the valve seat **26***b* and are identical in inside and outside diameters to each other, is fixedly mounted on an outer circumferential marginal edge of the valve holder **26**.

As shown in FIG. **10**, the intermediate sheet **29** has a pair of diametrically opposite sectorial recesses **29***a* defined therein and opening toward a free inner circumferential edge thereof. The recesses **29***a* serve as a variable orifice to allow the auxiliary valve **28** to flex axially for thereby generating a damping force when the piston **23** moves in a low-speed range. The damping force characteristics of the damper in low- and medium-speed ranges can easily be varied by adjusting the size and number of the recesses **29***a*.

The upper valve **31** has a plurality of slits **31***a* defined at spaced intervals in an outer circumferential edge thereof. The upper valve **31** is held against an outer circumferential portion of the piston **23** by the valve holder **26** which is resiliently biased by the spring **27**. The valve holder **26**, the auxiliary valve **28**, the intermediate sheet **29**, the main valve **30**, and the upper valve **31** jointly constitute a check valve. When the check valve is closed, the slits **31***a* of the upper valve **31** serve as a fixed orifice for generating a damping force when the piston **23** moves in the low-speed range.

When the piston **23** moves in an expansion stroke in the low-speed range in which the oil pressure in the oil chamber S1 is slightly higher than that in the oil chamber S2, the auxiliary valve **28** does not open as shown in FIG. **11**(*a*). At this time, the generated damping force gradually increases from a region encircled by a dotted-line circle in FIG. **11**(*b*), depending on the opening of the fixed orifice composed of the slits **31***a* in the upper valve **31**.

As the piston speed approaches a medium-speed range, working oil flowing into the oil passages **26***a* forces free-edge portions of the auxiliary valve **28** to flex axially into the corresponding recesses **29***a* in the intermediate sheet **29**, as shown in FIG. **12**(*a*), producing a damping oil path as indicated by the arrow in FIG. **12**(*a*). Depending on the

piston speed, the cross-sectional area of the damping oil path varies (increases), making the characteristic curve of the generated damping force smooth without inflections in a region encircled by a dotted-line circle in FIG. **12**(*b*).

When the piston **23** moves in a high-speed range, the intermediate sheet **29** is flexed in its entirety under the pressure of working oil flowing into the oil passages **26***a*, opening the entire check valve, as shown in FIG. **13**(*a*). Now, a damping force path as indicated by the arrows in FIG. **13**(*a*) is formed between the check valve and the valve seat **26***b* in its entirety. At this time, since the valve seat **26***b* is of a circular shape, the generated damping force gradually increases, without a sharp rise, in a region encircled by a dotted-line circle in FIG. **13**(*b*). In a compression stroke, the spring **27** flexes, and a fixed orifice is defined around the valve holder **26** which serves as part of the check valve. The valve structure shown in FIG. **9** in which the valves **28, 29, 30, 31** are supported at their outer circumferential edges on the piston **23** offers the same advantages as those of the valve structures according to the first and second embodiment in which the valves are supported at their inner circumferential edges.

In the third embodiment of the present invention, as described above, the intermediate sheet sandwiched between the main valve and the auxiliary valve is of the same overall inside and outside diameters between recesses as those of the main valve and the auxiliary valve, and has sectorial recesses. The intermediate sheet is effective to provide substantially linear damping force characteristics from the medium-speed range to the high-speed range, and damping force characteristics determined by the valve seat in the high-speed range. It is also easy to grasp the correlation between the shape of the intermediate sheet and the generated damping force depending on the number, shape, angle, and depth of the recesses in the intermediate sheet.

Although there have been described what are at present considered to be the preferred embodiments of the invention, it will be understood that the invention may be embodied in other specific forms without departing from the essential characteristics thereof. The present embodiments are therefore to be considered in all respects as illustrative, and not restrictive. The scope of the invention is indicated by the appended claims rather than by the foregoing description.

What is claimed is:

1. A hydraulic damper, comprising:

a cylinder;

a piston rod axially movable in said cylinder;

a dividing member mounted on an end of the piston rod and dividing an interior space of the cylinder into first and second fluid chambers, one on each side of the dividing member, the dividing member having passages to permit the passage of fluid between the first and second chambers and a valve seat;

an annular, resilient valve assembly having inner and outer circumferential edges, one of the circumferential edges being fixed with respect to the dividing member, the other of the circumferential edges being free and seatable on said valve seat;

the valve assembly comprising an annular auxiliary resilient valve, an annular intermediate resilient sheet and an annular resilient main valve, said annular auxiliary resilient valve, said annular intermediate resilient sheet and said annular resilient main valve being stacked in the order named from said valve seat;

the free circumferential edge of the annular intermediate resilient sheet being interrupted by at least one recess which opens from the free circumferential edge;

the annular auxiliary resilient valve, the annular intermediate resilient sheet and the annular resilient main valve having identical overall inside and outside circumferential edge diameters except at the recess;

whereby said annular resilient valve assembly can flex away from the valve seat depending on a differential pressure between the first and second chambers to produce a damping oil path for applying a damping force against axial movement of said piston rod with respect to said cylinder, the recess in the free circumferential edge of the annular intermediate resilient sheet allowing a portion of the annular auxiliary resilient valve to flex away from the valve seat to produce a damping oil path independently of the remainder of the valve assembly.

2. A valve structure according to claim **1**, wherein said valve seat is of a circular shape.

3. A valve structure according to claim **1**, wherein said recess is of a sectorial shape.

4. A valve structure according to claim **1**, wherein said recess is of a semicircular shape.

5. A valve structure according to claim **1**, wherein said recess is of a U shape.

6. The hydraulic damper of claim **1**, wherein the inner circumferential edge of the valve assembly is fixed with respect to the divider and the outer circumferential edge is free with respect to the divider.

7. The hydraulic damper of claim **1**, wherein the outer circumferential edge of the valve assembly is fixed with respect to the divider and the inner circumferential edge is free with respect to the divider.

8. The hydraulic damper of claim **7**, wherein the dividing member further comprises a valve guide enclosing the outer circumferential edge of the valve assembly, a piston provided with at least one fluid passage, a circular valve holder carried within the valve guide and having at least one fluid passage, the valve holder being provided with the valve seat and the valve assembly being disposed between the piston and the valve holder.

9. The hydraulic damper of claim **8**, wherein the valve assembly further comprises an annular, resilient upper valve stacked above the main valve with respect to the valve seat, the upper valve having an outer circumference that is interrupted with at least one slit, the valve assembly being carried by the piston.

10. The hydraulic damper of claim **9**, wherein the valve holder is movable with respect to the valve guide, further comprising a spring to urge the valve holder toward the piston.

11. A hydraulic damper, comprising:

a cylinder;

a piston rod axially movable in said cylinder;

a dividing member mounted on an end of the piston rod and dividing an interior space of the cylinder into first and second fluid chambers, one on each side of the dividing member, the dividing member having passages to permit the passage of fluid between the first and second chambers and a valve seat;

an annular, resilient valve assembly having inner and outer circumferential edges, one of the circumferential edges being fixed with respect to the dividing member, the other of the circumferential edges being free and seatable on said valve seat;

the valve assembly comprising an annular auxiliary resilient valve, and an annular resilient main valve, said annular auxiliary resilient valve, and said annular resil-

ient main valve being stacked in the order named from said valve seat;

the free circumferential edge of the annular resilient main valve being interrupted by at least one recess which opens from the free circumferential edge;

the annular auxiliary resilient valve and the annular resilient main valve having identical overall inside and outside circumferential edge diameters except at the recess;

whereby said annular resilient valve assembly can flex away from the valve seat depending on a differential pressure between the first and second chambers to produce a damping oil path for applying a damping force against axial movement of said piston rod with respect to said cylinder, the recess in the free circumferential edge of the annular resilient main valve allowing a portion of the annular auxiliary resilient valve to flex away from the valve seat to produce a damping oil path independently of the remainder of the valve assembly.

12. The hydraulic damper of claim **11**, wherein the valve seat is circular.

13. The hydraulic damper of claim **11**, wherein the recess is of a sectorial shape.

14. The hydraulic damper of claim **11**, wherein the recess is of a semicircular shape.

15. The hydraulic damper of claim **11**, wherein the recess is of a U shape.

16. A valve assembly for use in a hydraulic damper which comprises a cylinder, a piston rod axially movable in said cylinder, and a dividing member mounted on an end of the piston rod and dividing an interior space of the cylinder into first and second fluid chambers, one on each side of the dividing member, the dividing member having passages to permit the passage of fluid between the first and second chambers and a valve seat, the valve assembly being annular and resilient and comprising:

inner and outer circumferential edges, one of the circumferential edges to be fixed with respect to the dividing member, the other of the circumferential edges to be free and seatable on said valve seat;

the valve assembly further comprising an annular auxiliary resilient valve, an annular intermediate resilient sheet and an annular resilient main valve, said annular auxiliary resilient valve, said annular intermediate resilient sheet and said annular resilient main valve to be stacked in the order named from said valve seat;

the free circumferential edge of the annular intermediate resilient sheet being interrupted by at least one recess which opens from the free circumferential edge;

the annular auxiliary resilient valve, the annular intermediate resilient sheet and the annular resilient main valve having identical overall inside and outside circumferential edge diameters except at the recess;

whereby said annular resilient valve assembly can flex away from the valve seat depending on a differential pressure between the first and second chambers to produce a damping oil path for applying a damping force against axial movement of said piston rod with respect to said cylinder, the recess in the free circumferential edge of the annular intermediate resilient sheet allowing a portion of the annular auxiliary resilient valve to flex away from the valve seat to produce a damping oil path independently of the remainder of the valve assembly.

17. A valve assembly for use in a hydraulic damper which comprises a cylinder, a piston rod axially movable in said cylinder, and a dividing member mounted on an end of the piston rod and dividing an interior space of the cylinder into first and second fluid chambers, one on each side of the dividing member, the dividing member having passages to permit the passage of fluid between the first and second chambers and a valve seat, the valve assembly being annular and resilient and comprising:

inner and outer circumferential edges, one of the circumferential edges to be fixed with respect to the dividing member, the other of the circumferential edges to be free and seatable on said valve seat;

the valve assembly further comprising an annular auxiliary resilient valve, and an annular resilient main valve, said annular auxiliary resilient valve, and said annular resilient main valve to be stacked in the order named from said valve seat;

the free circumferential edge of the annular resilient main valve being interrupted by at least one recess which opens from the free circumferential edge;

the annular auxiliary resilient valve and the annular resilient main valve having identical overall inside and outside circumferential edge diameters except at the recess;

whereby said annular resilient valve assembly can flex away from the valve seat depending on a differential pressure between the first and second chambers to produce a damping oil path for applying a damping force against axial movement of said piston rod with respect to said cylinder, the recess in the free circumferential edge of the annular resilient main valve allowing a portion of the annular auxiliary resilient valve to flex away from the valve seat to produce a damping oil path independently of the remainder of the valve assembly.

* * * * *

United States Patent [19]

Yamaura et al.

[11] **Patent Number: 4,964,493**

[45] **Date of Patent: Oct. 23, 1990**

[54] **SHOCK ABSORBER WITH VARIABLE DAMPING CHARACTERISTICS DEPENDING UPON STROKE SPEED**

[75] Inventors: **Tamotsu Yamaura; Fumiyuki Yamaoka; Mitsuo Sasaki,** all of Kanagawa, Japan

[73] Assignee: **Atsugi Motor Parts Company, Limited, Kanagawa, Japan**

[21] Appl. No.: **333,337**

[22] Filed: **Apr. 5, 1989**

[30] **Foreign Application Priority Data**

Apr. 6, 1988	[JP]	Japan	63-46494[U]
Apr. 6, 1988	[JP]	Japan	63-46495[U]
Apr. 7, 1988	[JP]	Japan	63-46966[U]
Jan. 12, 1989	[JP]	Japan	1-2578[U]

[51] **Int. Cl.^{5}** **F16F 9/348**

[52] **U.S. Cl.** **188/322.15;** 188/280

[58] **Field of Search** 188/280, 320, 322.14, 188/322.15

[56] **References Cited**

U.S. PATENT DOCUMENTS

4,076,276	2/1978	Wijnhoven et al.	188/322.15 X
4,352,417	10/1982	Stinson	188/322.15
4,460,074	7/1984	Müller et al.	188/322.15 X
4,610,332	9/1986	Mourray	188/322.15

FOREIGN PATENT DOCUMENTS

3590858	9/1958	Australia .
4668764	1/1966	Australia .
310966	9/1967	Australia .
2622777	1/1979	Australia .
3109122	9/1982	Fed. Rep. of Germany .
1545406	9/1968	France .
61-47134	3/1986	Japan .

Primary Examiner—Duane A. Reger

Attorney, Agent, or Firm—Bachman & LaPointe

[57] **ABSTRACT**

A shock absorber is provided a piston stroke dependent variable shock absorbing characteristics by means of a first and second flow restriction valves associated with a piston. The first and second flow restriction valves are arranged in a fluid path for communicating first and second working chambers defined within a shock absorber cylindrical housing in series. The first flow restriction valve is associated with an orifice forming a part of the fluid path and has variable flow restriction rate in response to a pressure difference between the first and second chambers greater. The second flow restriction means is disposed in tandem fashion with the first flow restriction valve for providing a predetermined constant flow restriction rate in response to the pressure difference smaller than a predetermined criterion and providing a variable flow restriction as a function of the piston stroke speed in response to the pressure difference greater than the predetermined criterion.

19 Claims, 10 Drawing Sheets

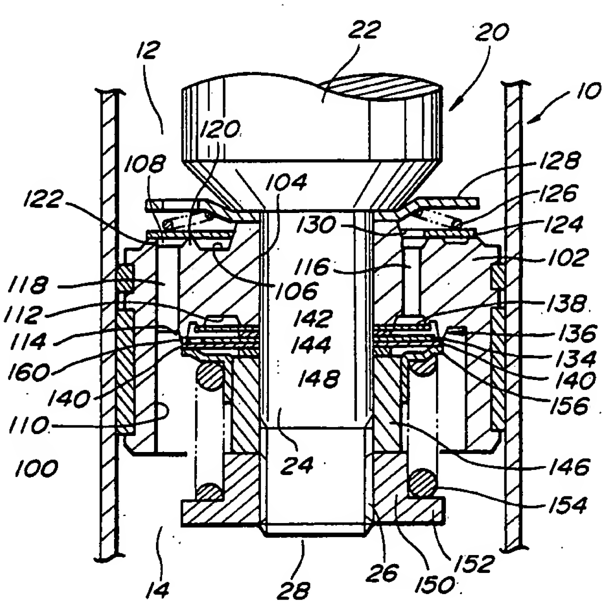


FIG. 1

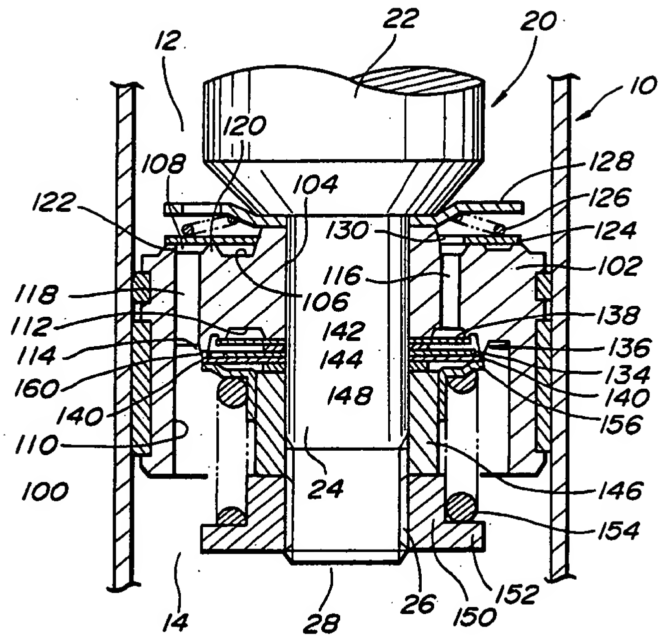

12
22
20
10
120
108
104
128
122
126
130
124
116
102
118
106
138
112
142
136
114
144
134
160
148
140
140
156
110
146
100
24
154
152
26
14
28
150

FIG. 2

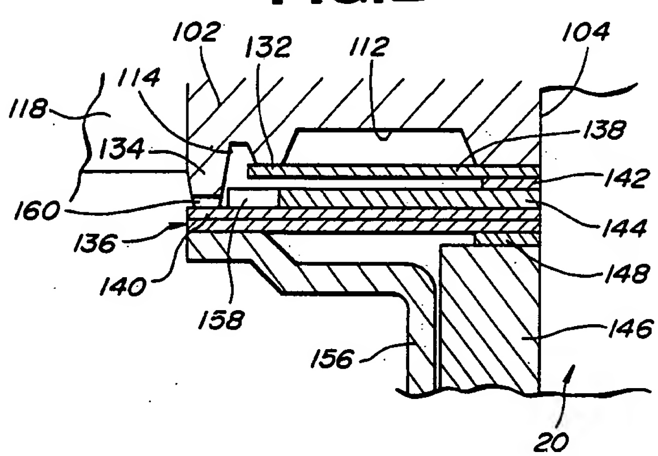

102
132
112
104
118
114
138
134
142
160
144
136
148
140
158
146
156
20

FIG. 3

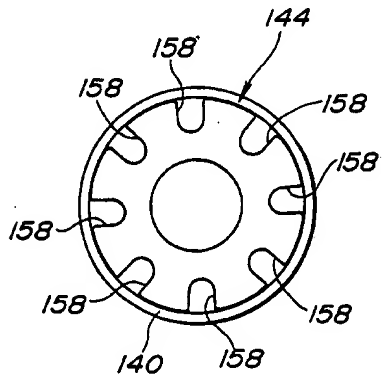


FIG. 4

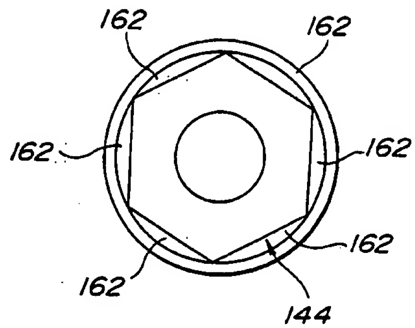


FIG. 5

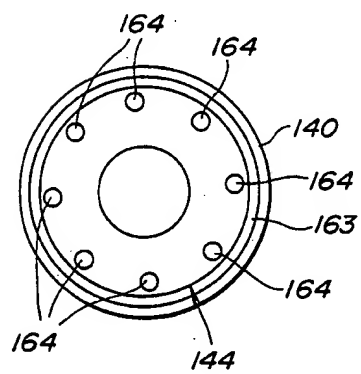


FIG. 6

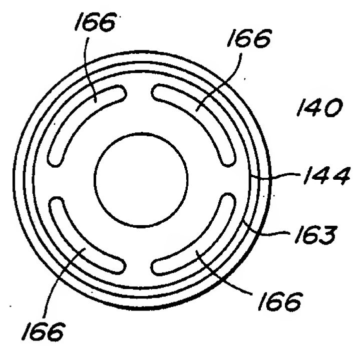


FIG. 7

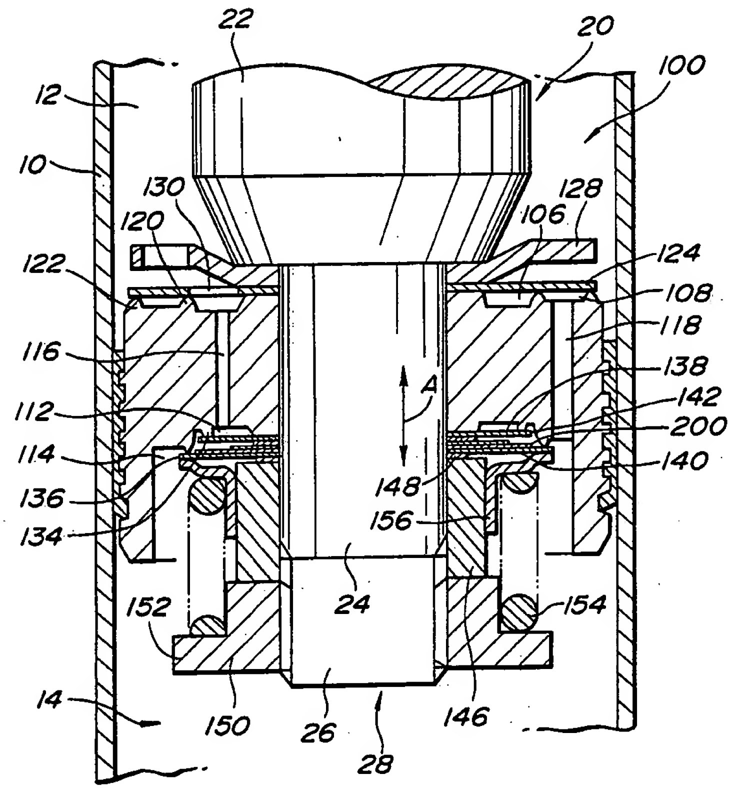

22
20
100
12
10
130
128
106
120
124
122
108
118
116
A
138
142
112
200
114
140
148
136
156
134
24
152
154
14
150
26
146
28

FIG. 8

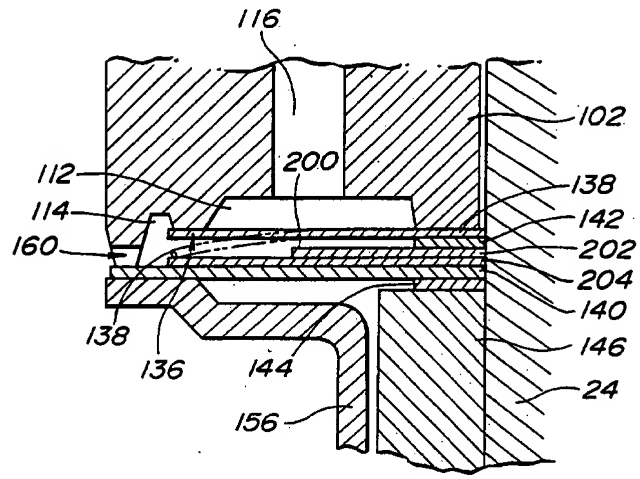

116
102
200
112
138
114
142
160
202
204
140
138
146
136
144
24
156

FIG. 9

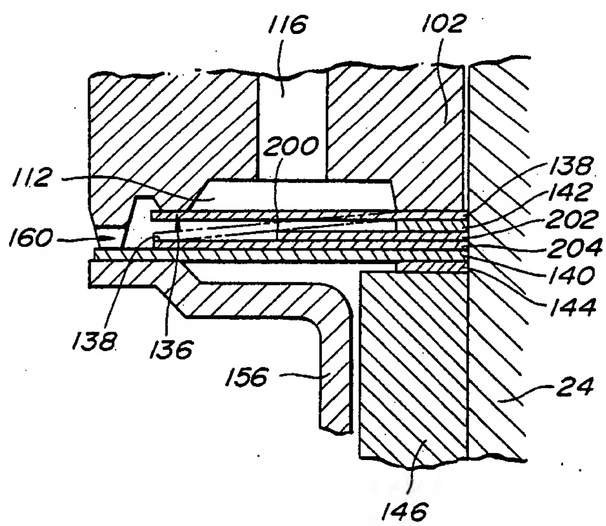

116
102
200
138
112
142
202
160
204
140
144
138
136
156
24
146

FIG. 10

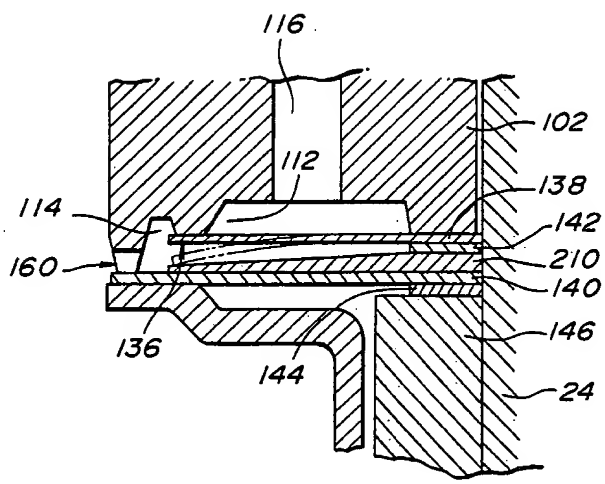


FIG.11

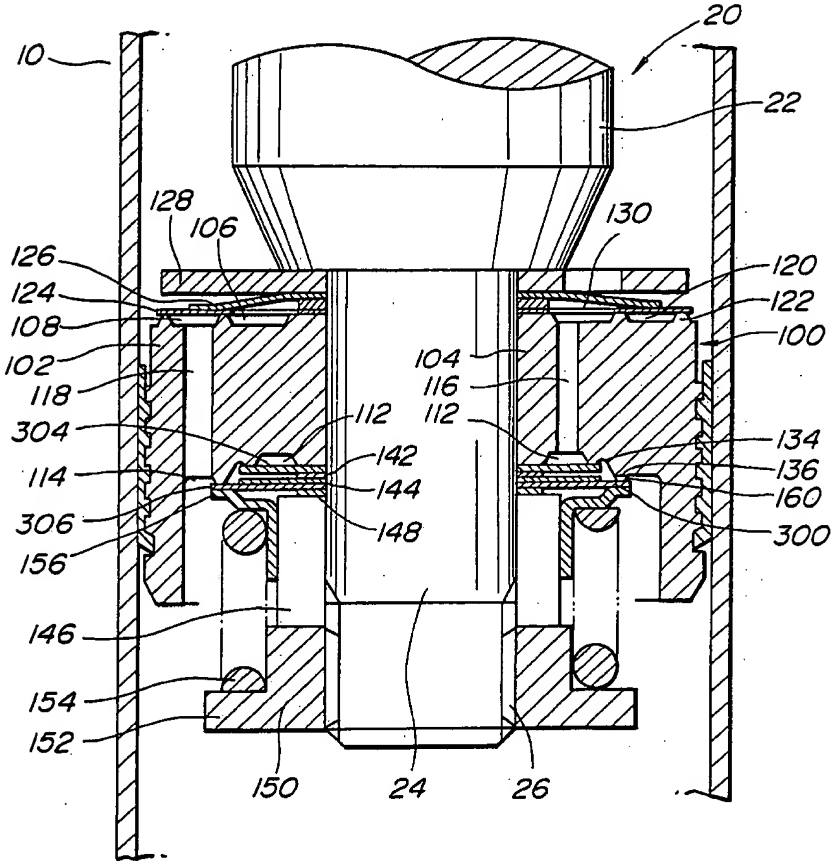

10
20
22
128
106
130
126
120
124
122
108
100
102
104
118
116
112
112
304
134
142
136
114
144
160
306
148
300
156
146
154
152
150
24
26

FIG.12

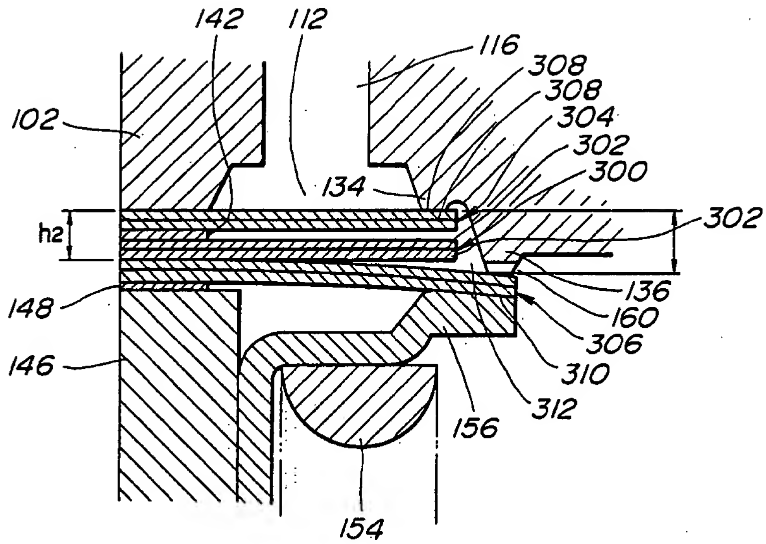

142
112
116
308
308
304
302
300
102
134
h2
302
136
160
148
306
146
310
312
156
154

FIG.13

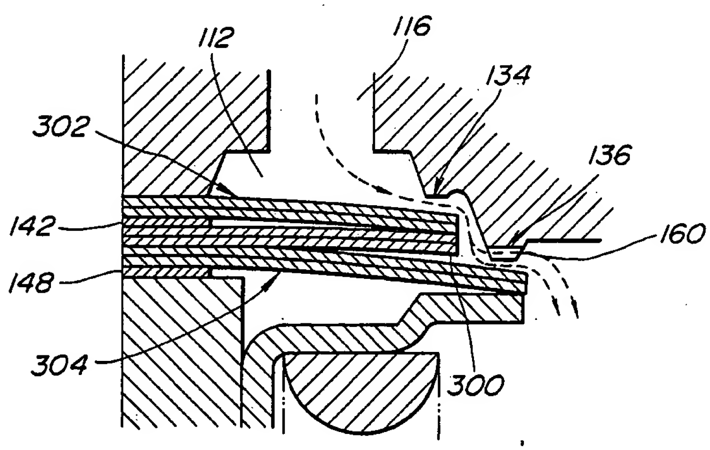

112
116
134
302
136
142
160
148
304
300

FIG.14

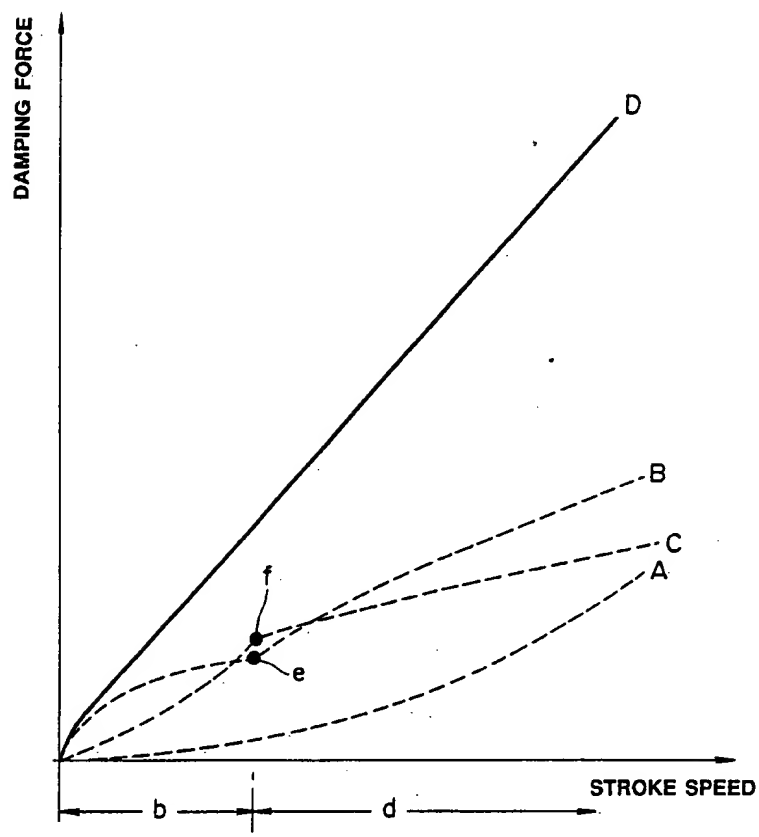
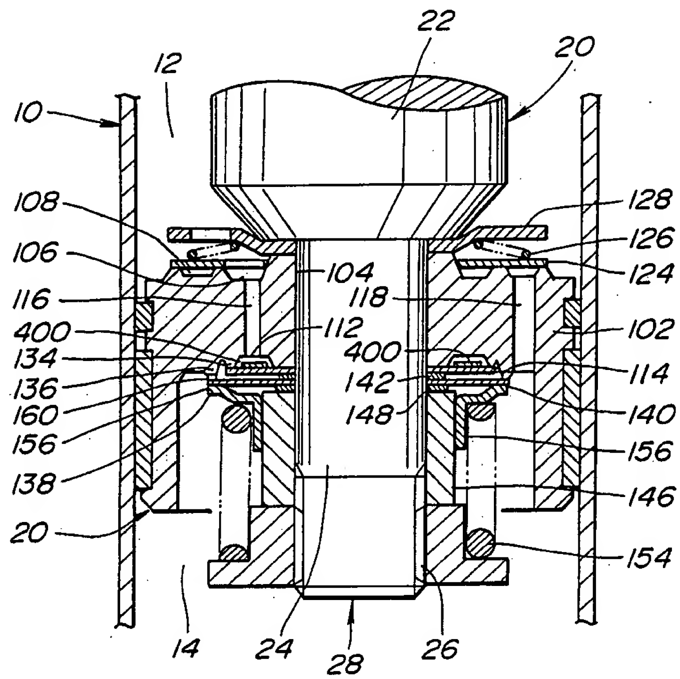
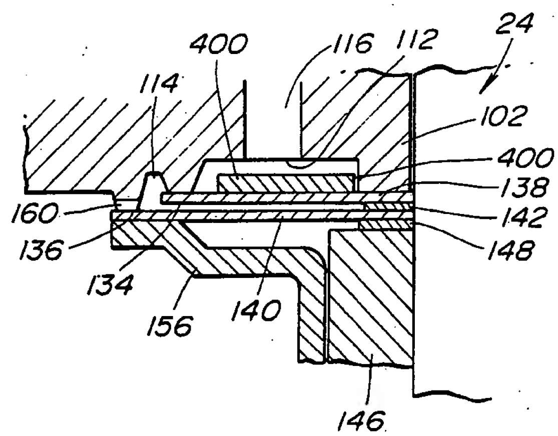


FIG.15



22
20
12
10
108
106
116
400
134
136
160
156
138
20
128
126
124
104
118
112
400
142
148
102
114
140
156
146
154
14
24
28
26

FIG.16



400
116
112
24
114
102
400
138
160
142
136
148
134
156
140
146

FIG.17

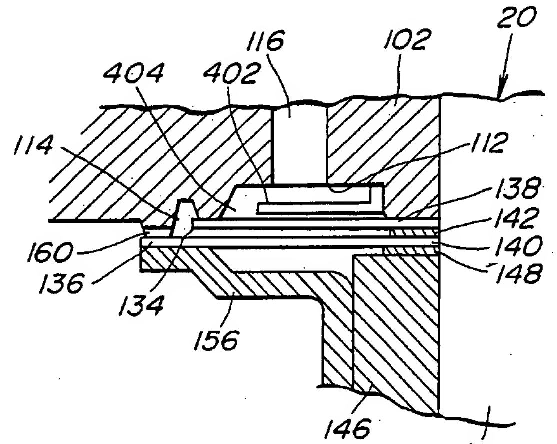


FIG.18

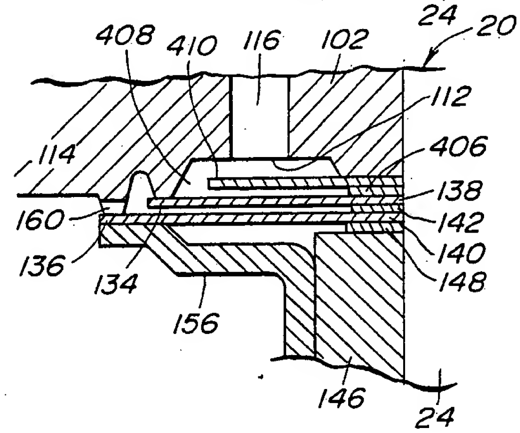
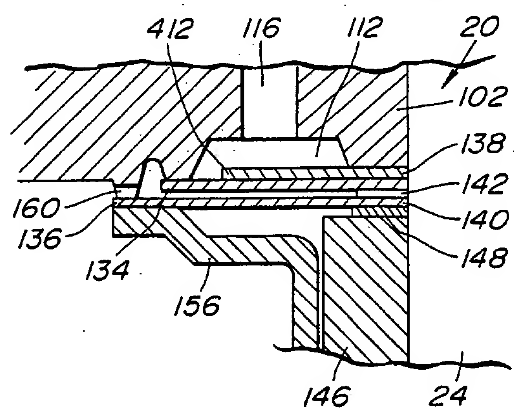


FIG.19



SHOCK ABSORBER WITH VARIABLE DAMPING CHARACTERISTICS DEPENDING UPON STROKE SPEED

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates generally to a hydraulic shock absorber, such as for automotive suspension system. More specifically, the invention relates to a valve construction to be employed in the hydraulic shock absorber for achieving piston stroke speed dependent variable damping characteristics.

2. Description of the Background Art

It should be appreciated that in automotive suspension system a shock absorber is required to successfully absorb vibration energy which causes vibration in the vehicle body in order to provide for riding comfort of the vehicle. On the other hand, the shock absorber has to damp vibration for suppressing relative displacement between the vehicle body and a suspension member so as to suppress attitude change for driving stability.

It should also be appreciated that vibration energy generated in a road wheel due to uneveness generally influences the riding comfort of the vehicle and has relatively small magnitude and high frequency to cause small magnitude and high speed piston stroke in the shock absorber. On the other hand, vibration energy induced in the vehicle body generally influences the driving stability for causing attitude change, such as pitching, rolling and so forth, and has relatively great magnitude and low frequency to cause great magnitude and low speed piston stroke.

In order to provide both of the riding comfort and driving stability, it is therefore required to absorb high frequency vibration and to damp low frequency vibration. In the prior art, there have been proposed various shock absorbers which attempted to provide piston stroke dependent damping characteristics for accomplishing both of the aforementioned tasks.

For example, Japanese Utility Model First (unexamined) Publication (Jikkai) Showa 61-47134 discloses a shock absorber with a multi-stage valve assembly employed in a shock absorber piston. In the shown construction of the valve assembly, first stage and second stage disc valves are arranged in series or in tandem fashion with respect to a fluid path for fluid communication between upper and lower working chambers defined in a shock absorber cylinder. The first stage valve is designed to respond to the lower pressure to be exerted thereonto to open for fluid communication therethrough. On the other hand, the second stage valve is designed to respond to the higher pressure than the first stage valve to open for fluid communication therethrough. The second stage valve also defines a constant orifice or orifices having a predetermined fixed fluid path area for constantly permitting fluid flow therethrough in a limited flow rate.

With the construction set forth above, in a response to the low speed piston stroke which creates a smaller pressure difference between the upper and lower working chamber and thus small pressure to exert on the first stage and second stage valves. The first stage valve is responsive to this small pressure to establish fluid communication between the upper and lower fluid chambers. Therefore, the working fluid flows through the gap formed in the first stage valve and the constant orifice. In such case, since the path area is limited to be small, substantially great flow restriction for the working fluid is provided to generate great damping force to damp the vibration induced in the vehicle body. On the other hand, in response to high frequency piston stroke which creates greater pressure difference between the upper and lower working chambers, both first stage and second stage valves are open to provide increased fluid path area to produce smaller damping force. Therefore, the vibration energy input from the road wheel can be absorbed to avoid a rough ride.

In such a valve construction, the first stage valve is driven to deform at substantially higher frequency than that of the second stage valve. Therefore, the first stage valve may have shorter life in comparison with that of the second stage valve. Particularly, when the first stage valve has constant resilient characteristics, it may be subject substantially high stress to further shorten the life.

On the other hand, at a pressure difference greater than the pressure relief point of the second stage valve, the damping characteristics at a flow restriction orifice extending through the shock absorber piston varies at a rate proportional to two power of the piston stroke speed. On the other hand, at the same time, the damping characteristics at the first and second stage valves varies at a rate proportional to two over three power of the piston stroke speed. As a result, variation characteristics of overall damping characteristics of the shock absorber cannot be linear to cause difficulty of setting desired variation characteristics.

SUMMARY OF THE INVENTION

Therefore, it is an object of the present invention to provide a hydraulic shock absorber which has a linear damping characteristics in relation to a piston stroke.

Another object of the invention is to provide a hydraulic shock absorber which can avoid fatigue of valve member for generating damping force and thus expand the life of the shock absorber.

In order to accomplish the aforementioned and other objects, a shock absorber, according to the present invention, has piston stroke dependent variable shock absorbing characteristics by means of first and second flow restriction valves associated with a piston. The first and second flow restriction valves are arranged in a fluid path for communicating first and second working chambers defined within a shock absorber cylindrical housing in series. The first flow restriction valve is associated with an orifice forming a part of the fluid path and has variable flow restriction rate in response to a pressure difference between the first and second chambers. The second flow restriction means is disposed in tandem fashion with the first flow restriction valve for providing a predetermined constant flow restriction rate in response to the pressure difference smaller than a predetermined criterion and providing a variable flow restriction as a function of the piston stroke speed in response to the pressure difference greater than the predetermined criterion.

According to one aspect of the invention, a flow restriction valve unit in a hydraulic shock absorber, which valve unit is provided in one of a piston assembly or a bottom valve assembly and associated with a fluid path for communication between first and second fluid chambers, comprises:

a flow restricting orifice forming a part of the fluid path for permitting fluid flow therethrough at a first limited flow rate;

a first upstream valve associated with the flow restricting orifice and normally closing one end of the orifice for blocking fluid communication between the first and second fluid chambers, and being responsive to fluid pressure difference between the first and second fluid chamber of greater magnitude than a predetermined first magnitude to open the one end of the orifice to establish fluid communication between the first and second fluid path for permitting fluid flow from the first fluid chamber to the second fluid flow chamber; and

a second downstream valve associated with the first upstream valve and arranged downstream of the first upstream valve in series with the latter with respect to the fluid flow from the first fluid chamber to the second fluid chamber, the second downstream valve having a predetermined constant flow path area of flow restriction path for communication between the downstream of the first upstream valve and the second fluid chamber, and being responsive to the fluid pressure difference between the downstream of the first upstream valve and the second fluid chamber to be greater than a second predetermined magnitude to increase the fluid path area,

the first upstream and second downstream valves being so cooperated to provide linear variation of damping force in accordance with variation speed of fluid pressure difference.

According to another aspect of the invention, a hydraulic shock absorber disposed between relatively movable first and second objects for absorbing vibration energy which causes relative movement between the first and second objects, comprising:

a cylinder tube defining an internal space and connected to the first object for movement therewith;

a piston assembly disposed within the internal space of the cylinder tube for defining first and second fluid chambers therein, the piston being connected to the second objects by means of a piston rod for movement therewith;

a flow restriction valve unit provided in the piston assembly associated with a fluid path for communication between first and second fluid chambers, the valve unit comprising:

- a flow restricting orifice forming a part of the fluid path for permitting fluid flow therethrough at a first limited flow rate;
- a first upstream valve associated with the flow restricting orifice and normally closing one end of the orifice for blocking fluid communication between the first and second fluid chambers, and being responsive to fluid pressure difference between the first and second fluid chamber of greater magnitude than a predetermined first magnitude to open the one end of the orifice to establish fluid communication between the first and second fluid path for permitting fluid flow from the first fluid chamber to the second fluid flow chamber; and
- a second downstream valve associated with the first upstream valve and arranged downstream of the first upstream valve in series with the latter with respect to the fluid flow from the first fluid chamber to the second fluid chamber, the second downstream valve having a predetermined constant flow path area of flow restriction path for communication between the downstream of the first upstream valve and the second fluid chamber, and being responsive to the fluid pressure difference between the downstream of the first upstream valve and the second fluid chamber to be greater than a second predetermined magnitude to increase the fluid path area,
- the first upstream and second downstream valves being so cooperated to provide linear variation of damping force in accordance with variation speed of fluid pressure difference.

Preferably, the first upstream valve has a valve member capable of shifting in accordance with increasing the fluid pressure in the first fluid chamber, the valve member, being cooperated with a stopper means which limits the shifting range of the valve member. The valve member may comprise a resiliently deformable disc, and the stopper means comprises a disc shaped member having substantially the same diameter to that of the valve member so that at least the circumferential edge portion of the valve member seats thereon at a predetermined magnitude of resilient deformation, which stopper means defines a clearance to permit fluid flow from the first upstream valve to the second downstream valve through the clearance. The disc shaped member may comprise a first smaller diameter disc and a second larger diameter disc which has essentially the same diameter to the valve member, the second disc being oriented to limit deformation magnitude by contacting with the circumferential edge of the valve member and the first disc being oriented to limit deformation at the intermediate portion of the valve member. Alternatively, the disc shaped member is a resiliently deformable. In the further alternative embodiment, the disc shaped member may comprise a plurality of resiliently deformable discs laminated to each other.

The hydraulic shock absorber may further comprise a plate like member disposed between the first upstream valve and the discharge outlet of the orifice for receiving working fluid flow and distributing the uniform fluid pressure to the first upstream valve. In the alternative embodiment, the pressure difference between the first and second fluid chamber is created by stroke of the piston assembly.

In the preferred constriction, the orifice provides damping characteristics which vary at a rate proportional to two power of the stroke speed of the piston assembly; the first upstream valve provides damping characteristics which vary at a rate proportional to two over three power of the stroke speed of the piston assembly; and the second downstream valve provides damping characteristics in response to the stroke speed of the piston assembly lower than or equal to a predetermined piston speed criterion, varying at a rate proportional to two power of the stroke speed of the piston assembly, and in response to the stroke speed of the piston assembly higher than the piston stroke criterion, varying at a rate proportional to two over three power of the stroke speed of the piston assembly.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be understood more fully from the detailed description given herebelow and from the accompanying drawings of the preferred embodiments of the present invention, which, however, should not be taken to limit the invention to the specific embodiment or embodiments, but are for explanation and understanding only.

In the drawings:

FIG. 1 is a section of a major part including a piston assembly of the first embodiment of a hydraulic shock absorber according to the present invention;

FIG. 2 is an enlarged section of a flow restricting valve unit employed in the first embodiment of the hydraulic shock absorber according to the invention;

FIGS. 3 to 6 show examples of stopper plates to be employed in the first embodiment of the hydraulic shock absorber of FIG. 1;

FIG. 7 is a section of a piston assembly in the second embodiment of a hydraulic shock absorber according to the present invention;

FIGS. 8 and 9 are enlarged sections of valve units employed in the second embodiment of the hydraulic shock absorber in different mode positions;

FIG. 10 is an enlarged section of a modification of valve unit to be employed in the second embodiment of the hydraulic shock absorber of FIG. 7;

FIG. 11 is a section of a piston assembly in the third embodiment of a hydraulic shock absorber according to the present invention;

FIGS. 12 and 13 are enlarged sections of valve units employed in the third embodiment of the hydraulic shock absorber in different mode positions;

FIG. 14 is a graph shocking variation of damping force to be generated at various piston stroke speed;

FIG. 15 is a section of a piston assembly in the fourth embodiment of a hydraulic shock absorber according to the present invention;

FIG. 16 is an enlarged section of a valve unit employed in the piston assembly of the fourth embodiment of the hydraulic shock absorber of FIG. 15; and

FIGS. 17 to 19 are enlarged sections showing variation of the valve unit to be employed in the fourth embodiment of the hydraulic shock absorber of FIG. 16.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings, particularly to FIG. 1, the first embodiment of a hydraulic shock absorber, according to the present invention, employs a piston assembly 100 disposed within a cylinder tube 10 which forms a shock absorber housing filled with a hydraulic working fluid. The piston assembly 100 is disposed within the internal space of the cylinder tube 10 in order to divide the internal space into an upper and a lower fluid chamber 12 and 14. The piston assembly 100 is mounted at the lower end of a piston rod 20 which has an upper end extending through the upper end of the cylinder tube 10. In the practical use, the shock absorber is disposed between upper and lower objects which are movable relative to each other, so as to absorb vibration energy for causing relative movement of the upper and lower objects. Though the discussion herein is given in terms of vertical installation of the shock absorber, it does not specify the direction of installation of the shock absorber. Namely, the shock absorber can be installed horizontally so as to absorb vibration energy between horizontally arranged two objects.

When the shock absorber is vertically installed as hereinbelow discussed, the piston rod 20 is connected to the upper object for moving the piston assembly 100, and the lower end of the cylinder tube 10 is connected to the lower object. In case that the shock absorber is employed in an automotive suspension system, the piston rod 20 is connected to a vehicle body at the top end and the cylinder tube 10 is connected to a suspension member which rotatably supports a road wheel. Though the vertical position of the shock absorber hereinafter discussed places the piston rod extending upwardly, it is possible to alternate the position to extend the piston rod downwardly.

As seen from FIG. 1, the piston rod 20 has a greater diameter major section 22 and a smaller diameter lower end section 24. The lower end section 24 has a threaded lower end 26. The lower end section 24 with the threaded lower end 26 forms a piston assembly receptacle 28.

The piston assembly 100 has a piston body 102 as principle component of the piston assembly. The piston body 102 has a center hole 104, through which the lower end section 24 of the piston rod 20 extends. The piston body 102 has an upper surface facing the upper working chamber 12 and a lower surface facing the lower working chamber 14. Coaxially arranged inner and outer annular grooves 106 and 108 are formed on the upper surface of the piston body 102. On the other hand, a circular groove 110 is formed on the lower surface of the piston body 102. Coaxially arranged inner and outer annular grooves 112 and 114 are formed on the surface facing groove 110. The inner annular grooves 106 and 112 are vertically aligned to each other. One or more fluid flow orifices 116 are formed through the piston body in parallel to the axis of the piston rod 20 for communication between the inner grooves 106 and 112. Similarly, the outer grooves 108 and 114 are arranged in vertical alignment and communicated with one or more communication paths 118.

Along the inner and outer grooves 106 and 108 on the upper surface of the piston body 102, annular lands 120 and 122 are formed. The lands 120 and 122 have upper planar surfaces which serve as valve seats for seating an annular disc shaped check valve member 124. The check valve member 124 is biased toward the valve seat surfaces of the lands 120 and 122 by means of a coil form check spring 126 which is disposed between the check valve member 124 and a retainer disk 128. With this construction, the check valve member 124 closes the annular grooves 106 and 108. A plurality of through openings 130 are formed in circumferential alignment at an orientation corresponding to the radial position of the inner groove 106. Therefore, the inner groove 106 is normally in communication with the upper working chamber 12 via these through openings 130. On the other hand, the check valve member 124 has solid construction in the orientation corresponding to the outer annular groove 108. Therefore, the check valve member 124 normally blocks fluid communication and response to the fluid pressure in the lower working chamber 14 higher than that in the upper working chamber in a magnitude greater than a predetermined relief pressure overcoming the spring force of the check spring 126 to be shifted upwardly to establish fluid communication between the lower and upper working chambers 14 and 12. On the other hand, the check valve member 124 constantly blocks fluid flow from the upper working chamber 12 to the lower working chamber 14.

Lands 132 and 134 are formed along the circumferential edges of the inner and outer annular grooves 112 and 114. The lands 132 and 134 have lower end planar surfaces. A piston stroke speed dependent variable damping characteristics valve unit 136 is provided in opposition to the lower surfaces of the lands 132 and 134.

As shown in FIGS. 1 and 2, the valve unit 136 comprises an upper disc valve member 138 and a lower disc valve member 140. An annular washer 142 and a stopper plate 144 are disposed between the upper and lower disc valve members 138 and 140. These upper and lower disc valve members 138 and 140, the washer 142 and the stopper plate 144 forms the flow restriction valve unit 136 and are supported by an annular cylindrical collar 146 via disc shaped washer 148 on a retainer nut 150 which engages with the threaded lower end 26 of the piston rod 20.

The retainer nut 150 has a laterally extending flange 152, on which a lower end of an assist coil spring 154 is seated. The upper end of the assist coil spring 154 is seated on a spring seat member 156. The spring force of the assist spring 154 is thus exerted on the outer circumferential edge portion of the lower disc valve member 140 via the spring seat member 156 so that the outer circumferential edge of the lower disc valve member 140 is normally urged toward the lower planer surface of the outer annular land 134.

The upper and lower disc valve members 138 and 140 are formed of resiliently deformable material and provided spring characteristics for self-biasing the respective circumferential edge portions toward the opposing planer lower surfaces of the lands 134 and 132. The upper disc valve member 138 thus normally seats the outer circumferential edge portion on to the lower planar surface of the land 134 for closing the inner annular groove 112. Therefore, when the working fluid pressure in the upper working chamber 12 which is introduced into the inner annular groove 112 via the orifices 116 is not greater than the working fluid in the lower working chamber 14 in a magnitude greater than a predetermined pressure which is set pressure of the upper disc valve member 138 by the resiliency thereof, the inner annular groove 112 is held closed. Therefore, the fluid communication between the upper and lower working chambers 12 and 14 via the orifices 116 is blocked. The upper disc valve member 138 is overcome by the working fluid pressure in the upper working fluid chamber 12 greater than that in the lower working chamber 14 in a magnitude greater than the set pressure to form a gap between the lower end planer surface of the land 134 to permit fluid flow therethrough. The stopper plate 144 is formed with one or more cut outs 158 on the outer circumference. On the other hand, the outer annular land 136 is formed with one or more cut outs 160 which cooperate with the lower disc valve member 140 as seated on the seating surface of the land 136 to form a constant path area orifice for fluid communication. Therefore, when the gap is formed between the upper disc valve member 138 and the mating surface of the land 134, the working fluid flows through the cut outs 158 of the stopper plate 144 and the cut outs 160 of the lands 136. At this time, since the fluid flow path area at the gap between the upper disc valve member 138 and the land 134 is limited to restrict fluid flow, the gap serves as first stage orifice for generating damping force. On the other hand, the constant fluid flow path area orifices defined by the cut out 160 of the land 136 provides another flow restriction for the fluid flowing from the upper working chamber 12 to the lower working fluid chamber and thus serves as a second stage flow restriction orifice.

It should be noted that the resilience of the lower disc valve member 140 is cooperative with the spring force of the assist spring 154 to determine the set pressure of the lower disc valve member 140. The set pressure of the lower disc valve member 140 is determined to be greater than that of the upper disc valve member 138. In addition, as clearly seen from FIG. 2, the effective area of the first disc valve member 138, on which the working fluid pressure in the annular groove 112 is exerted, is set to be much greater than the effective area of the second disc valve member 140. Therefore, the upper disc valve member 138 is provided higher sensitivity of the working fluid pressure in the upper working chamber 12 greater than that in the lower working chamber 14, than the lower disc valve member 140.

As shown in FIGS. 2 and 3, the stopper plate 144 is formed of a rigid material and provided slightly smaller diameter than the inner diameter of the lower end of the annular land 136 to define a clearance therebetween. On the other hand, the cut outs 158 are formed into essentially U-shaped configuration and arranged along the outer circumference with a regular angular intervals.

In the practical operation, damping force for absorbing vibration energy in the piston stroke toward the lower working chamber with compressing the working fluid in the lower working chamber, is generated by flow restriction in the openings 118 which provide limited fluid flow path area and a spring force of the check spring 126.

On the other hand, in the piston stroke toward the upper working chamber 12 to cause compression of the working fluid in the upper working chamber, the valve unit 136 operates in two mutually different modes depending upon the piston stroke speed.

When the piston strokes at relatively low speed, variation rate of the working fluid pressure in the upper and lower fluid chambers 12 and 14 is held at relatively low rate. Therefore, speed of variation of pressure difference between the upper and lower working chamber 12 and 14 is maintained low. As a result, the working fluid force acting on the valve unit due to difference of the working fluid pressure in the upper and lower working chambers 12 and 14 increases at relatively low speed. When the fluid force is grown to be greater than the set pressure of the upper disc valve member 138, the upper disc valve member 138 is deformed to shift the circumferential edge portion away from the seating surface of the land 134 to form a gap therebetween to permit fluid flow therethrough. On the other hand, at this time, the working fluid pressure acting on the lower disc valve member 140 is maintained smaller than the set pressure thereof. Therefore, the working fluid flowing through the gap formed by deformation of the upper disc valve member 138 flows through the constant flow area orifices 160 formed through the land 136.

At this time, the damping characteristics at the first stage orifice formed by the deformation of the upper disc valve member 138, varies at a rate proportional to two over three power of the piston stroke speed. On the other hand, the sampling characteristics at the second stage orifice defined by the constant fluid path area orifices 160 varies at a rate proportional to two power of the piston stroke speed. Therefore, the damping force to be created in response to relatively low piston stroke speed becomes relatively greater.

On the other hand, when the piston strokes at relatively high speed, the pressure difference produced in the working fluid in the upper and lower working chambers 12 and 14 varies at greater rate than that in the low piston stroke speed set forth above. When the piston stroke speed is higher than a certain speed, the

working fluid pressure exerted on the lower disc valve 140 becomes greater than the set pressure of the lower disc valve member 140 to overcome the combined spring force of the valve member and the assist spring 154. Therefore, the lower disc valve member 140 is deformed to shift the circumferential edge portion away from the seating surface of the land 136 to form a gap therebetween. By this, the fluid flow area defined by the second stage orifice becomes greater than that in the low speed piston stroke.

At this time, the damping characteristics at the second stage orifice varies at a rate proportional to two over three of the piston stroke speed. Therefore, the flow restriction for the working fluid at the second stage orifice becomes smaller to provide smaller damping force in comparison with that in the low piston stroke speed mode.

It should be appreciated that the stopper plate 144 is made of rigid material so as not to cause the deformation in response to the fluid pressure acting thereon. As seen from FIG. 2, the stopper plate 144 is placed away from the upper disc valve member leaving clearance defined by the height of the washer 142. This construction limits deformation range of the upper disc valve member 138 to prevent the latter from causing excessive deformation. This assures expansion of the life of the upper disc valve member and thus expands durability of the shock absorber.

FIGS. 4 to 6 illustration of the modifications of configuration of the stopper member to be employed in the foregoing first embodiment. In FIG. 4, the stopper plate 144 is formed into an hexagonal configuration to form clearance 162 between the inner circumferential edge of the land 136 and the straight portion between the adjacent peaks. On the other hand, In FIG. 5, the stopper plate 144 is formed into a circular disc shaped configuration. The diameter of the circular disc shaped stopper plate 144 is set to be slightly smaller than the inner diameter of the inner circumferential edge of the land 136 in order to define an annular clearance 163. A plurality of circular through openings 164 are formed in the vicinity of the circumferential edge. The pressurized fluid flowing through these openings 164 acts on the corresponding portion of the lower disc valve member 140 so as to prevent the second disc valve member from sticking on the stopper plate. In FIG. 6, the stopper plate 144 is formed with a plurality of elongated arc shaped openings 166. The construction shown in FIG. 6 performs substantially the same effect to that performed by the construction of FIG. 5.

FIG. 7 shows the second construction of the hydraulic shock absorber according to the invention. In the discussion given herebelow, the common components to the former embodiment will be represented by the same reference numerals to the former embodiment and neglect the detailed discussions therefor.

In the shown embodiment, the construction of the piston assembly 100 is almost same as that of the former embodiment except for a stopper plate 200 which is modified from the stopper plate 144 in the first embodiment. Namely, in the shown embodiment, the stopper plate 200 comprises a pair of thinner plates 202 and 204. The plates 202 and 204 are formed in disc shaped configurations. The external diameter of the plate 204 is substantially equal to the diameter of the upper disc valve member 138. The plate 202 is provided smaller diameter than that of the plate 204. The plate 202 is placed on the plate 204.

In the construction set forth above, the valve unit 136 operates for producing damping force in response to the piston stroke toward the upper working chamber 12 in a manner substantially as illustrated with respect to the foregoing first embodiment of the invention. In the second embodiment of FIG. 7, when the piston stroke speed is relatively low, the magnitude of deformation of the upper disc valve member 138 is relatively small. In such case, the circumferential edge portion of the upper disc valve member 138 comes into contact with the plate 204 to be restricted from further deformation as shown by phantom line in FIG. 8. On the other hand, when the piston stroke speed is relatively high, whole body deformation is caused in the upper disc valve member 138 to cause deformation at the intermediate portion of the upper disc valve member. This causes expansion of the first stage orifice for permitting greater amount of working fluid. By the whole body deformation, the intermediate portion of the upper disc valve member 138 comes into contact with the plate 202 to be restricted from further deformation.

Such construction of valve unit may provide variable damping characteristics even in the low piston stroke speed mode of operation. Furthermore, limitation for deformation magnitude of the upper disc valve member 138 will successfully prevent the upper disc valve member from causing excessive deformation which may cause shortening of life of the upper disc valve member.

FIG. 10 shows modification of the aforementioned second embodiment of the valve unit to be employed in the second embodiment of the hydraulic shock absorber according to the invention. In this embodiment, the stopper plate 200 is replaced with a plate like member 210 with a slanted upper surface. As seen from FIG. 10, the plate line member 210 is formed in a disc shaped configuration and having the thinner circumferential edge and increases thickness toward the inside. With this construction, the deformation stroke of the upper disc valve member 138 is gradually and linearly increased toward the outer circumferential edge. With the shown construction, substantially the same effect for limiting deformation magnitude as that achieved by the embodiment of FIG. 7 can be achieved.

FIG. 11 shows the third embodiment of the hydraulic shock absorber according to the invention. Similarly to the former embodiment of FIG. 7, the shown embodiment has substantially the same structural components to that illustrated with respect to the first embodiment of FIG. 1. In order to avoid redundant discussion, common components between the shown embodiment and the first embodiment will be represented by the same reference numerals.

The third embodiment of the shock absorber is differentiated from the former embodiments of FIGS. 1 and 7 in the stopper plate 300 in the valve unit 136. The stopper plate 300 employed in the shown embodiment comprises a plurality of thin resiliently deformable discs 302 as particularly shown in FIGS. 12 and 13. The thin discs 302 are laminated to each other to form integral deformable disc. Also, the upper and lower disc valve members 304 and 306 respectively comprise a plurality of thin resiliently deformable discs 308 and 310. In the shown construction, respective upper and lower disc valve members 304 and 306 and the stopper plate 300 are formed by laminated pair of thin discs 308, 310 and 302. The resilient coefficient of respective thin discs 308, 310 and 302 are differentiated from each other so as to establish desired resilient coefficient in the laminated forms.

In the preferred construction, the resilient coefficient of the upper disc valve member 304 is set to be smallest among three laminated discs. On the other hand, the resilient coefficient of the stopper plate 302 may be set to be the greatest among three laminated plates.

As seen from FIGS. 11 to 13, the seating surfaces of the lands 134 and 136 are oriented at vertically shifted position to have a level difference in a distance h_1 in the vertical direction. In relation to the distance h_1, the thicknesses of the thin discs 308 and 302 and height of the washer 142 are so selected as to have the overall thickness h_2 as assembled smaller than the distance h_1. Therefore, the lower disc valve member 140 which has the circumferential edge portion seating on the seating surface of the land 136, is slightly deformed at the normal position as clearly shown in FIG. 12. This forms a clearance 312 between the lower disc valve member 306 and the stopper plate 300. The clearance 312 formed between the stopper plate 300 and the lower disc valve member 306 cancels the fluid pressure exerted on the stopper plate when the upper disc valve member 304 is deformed to form the first stage orifice. Therefore, the first stage orifice may not becomes excessively great to provide desired flow restriction for generating damping force.

Similarly to the foregoing first embodiment, the valve unit 136 performs two different mode flow restrictions for generating damping force.

When the piston strokes at relatively low speed in a piston stroke speed range as illustrated by the range b in FIG. 14, variation rate of the working fluid pressure in the upper and lower fluid chambers 12 and 14 is held at relatively low rate. Therefore, speed of variation of pressure difference between the upper and lower working chamber 12 and 14 is maintained low. As a result, the working fluid force acting on the valve unit due to difference of the working fluid pressure in the upper and lower working chambers 12 and 14 increases at relatively low speed. When the fluid force is grown to be greater than the set pressure of the upper disc valve member 304, the upper disc valve member 300 is deformed to shift the circumferential edge portion away from the seating surface of the land 134 to form a gap therebetween to permit fluid flow therethrough. At this position, variation of the damping force produced by the orifices is proportional to two power of the piston stroke speed, as illustrated by the broken line A of FIG. 14. Also, the variation of damping force to be produced at the first stage orifice is proportional to two over three power of the piston stroke speed as illustrated by broken line B in FIG. 14. On the other hand, at this time, the working fluid pressure acting on the lower disc valve member 140 is maintained smaller than the set pressure thereof. Therefore, the working fluid flowing through the gap formed by deformation of the upper disc valve member 138 flows through the constant flow area orifices 160 formed through the land 136. The variation of the damping force to be produced at the second stage orifice is proportional to two power of the piston stroke speed, as illustrated by the broken line C of FIG. 14. Since these orifices are arranged in tandem fashion or in series, overall damping force to be produce in this mode becomes almost liner as illustrated by the solid line D of FIG. 14.

In response to the piston stroke speed in a range of d of FIG. 14, the upper disc valve member 304 contacts with the stopper plate 300 to increase stiffness. Therefore, at a point e in FIG. 14, the stiffness of the upper disc valve member 304 is changed. Therefore, the variation of the damping force becomes proportional component which is proportional to two over three power of the piston stroke speed, and an initial component which is the damping force at the point e. On the other hand, at a point f of FIG. 14, the lower disc valve member 306 start deformation for forming gap between the seating surface of the land 136 and the circumferential edge portion of the lower disc valve member. After starting the deformation of the lower disc valve member 306, the variation characteristics of the damping force becomes proportional to tow over three power of the piston stroke speed. Therefore, as shown by the solid line D in FIG. 14, even at this high piston speed range, the damping force variation characteristics becomes linearly proportional to the piston stroke speed.

FIGS. 15 and 16 show the fourth embodiment of the shock absorber according to the present invention. The shown embodiment is illustrated in a form, in which the stopper plates in the former embodiments is not provided. Though the shown embodiment does not have the stopper plate, it is of course possible to provide the stopper plate in this embodiment.

The particular point of the shown embodiment resides in an annular disc plate 400. The annular disc plate 400 is provided within the inner annular groove 112 above the upper disc valve member 138. The annular disc plate 400 has a width to receive all working fluid discharged through the orifices 116. On the other hand, the outer diameter of the annular disc plate 400 is smaller than the diameter of the external circumferential edge of the inner annular groove 112 so that the working fluid introduced into the groove can flow through the gap therebetween.

This annular disc plate 400 prevents the working fluid from directly contacting with the upper disc valve member 138 to cause concentration at the position where the orifices 116 are formed. Since the concentration of the fluid pressure at the portion of the orifices will cause uneven deformation in the upper disc valve member to cause fluctuation of damping characteristics, it is desired to be avoided.

Similar effect for avoiding concentration of the fluid pressure at the particular section of the upper disc valve member 138 can be achieved by various construction of members. For example, in the construction of FIG. 17, the annular disc valve member 402 is fixed onto the inner peripheral edge of the groove 112 so that the annular disc valve member 402 is placed away from the upper disc valve member 138 to leave a clearance 404 therebetween. On the other hand, in the example of FIG. 18, a clearance 406 is provided by a washer 408 disposed between the annular disc 410 and the upper disc valve member 138. Though it is preferable to provide the annular disc in spaced apart relationship with the upper disc valve member 138, it is possible to provide the annular disc 412 together with the upper disc valve member 138 in engagement with the piston rod 24, as shown in FIG. 19.

While the present invention has been disclosed in terms of the preferred embodiment in order to facilitate better understanding of the invention, it should be appreciated that the invention can be embodied in various ways without departing from the principle of the invention. Therefore, the invention should be understood to include all possible embodiments and modifications to the shown embodiments which can be embodied with-

out departing from the principle of the invention set out in the appended claims.

What is claimed is:

1. A flow restriction valve unit in a hydraulic shock absorber, which valve unit is provided in one of a piston assembly or a bottom valve assembly and associated with a fluid path for communication between first and second fluid chambers, comprising:

a flow restricting orifice forming a part of said fluid path for permitting fluid flow therethrough at a first limited flow rate;

a first upstream valve associated with said flow restricting orifice and normally closing one end of said orifice for blocking fluid communication between said first and second fluid chambers, and being responsive to fluid pressure difference between said first and second fluid chamber greater magnitude than a predetermined first magnitude to open said one end of said orifice to establish fluid communication between said first and second fluid path for permitting fluid flow from said first fluid chamber to said second fluid flow chamber, said first upstream valve including a first resiliently deformable valve member which is normally biased in a direction for closing said one end of said orifice and deformable to increase fluid path area according to increasing fluid pressure difference between said first and second fluid chambers;

a second downstream valve associated with said first upstream valve and arranged downstream of said first upstream valve in series with the latter with respect to the fluid flow from said first fluid chamber to said second fluid chamber, said second downstream valve having a predetermined constant and minimum flow path area of flow restriction path for communication between the downstream of said first upstream valve and said second fluid chamber for providing a predetermined maximum magnitude of fluid flow restriction when the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be smaller than or equal to a second predetermined magnitude, and being responsive to the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be greater than a second predetermined magnitude to shift for increasing said fluid path area,

said second downstream valves being so cooperated with said first upstream valve as to provide linear variation of damping force in accordance with variation speed of fluid pressure difference;

said second downstream valve including a second resiliently deformable valve member which is normally biased in a direction for providing the minimum path area and deformable to increase fluid path area according to increasing fluid pressure difference between the downstream of said first upstream valve and second fluid chambers; and

a stopper means disposed between said first upstream and second downstream valves for defining a maximum deformation magnitude of said first valve member of said first upstream valve means.

2. A valve unit as set forth in claim 1, wherein said first upstream valve has a valve member capable of shifting in accordance with increasing of the fluid pressure in said first fluid chamber, said valve member being cooperated with a stopper means which limits shifting range of said valve member.

3. A valve unit as set forth in claim 2, wherein said valve member comprises a resiliently deformable disc, and said stopper means comprises a disc shaped member having substantially the same diameter to that of said valve member so that at least the circumferential edge portion of said valve member seats thereon at a predetermined magnitude of resilient deformation, which stopper means defines a clearance to permit fluid flow from said first upstream valve to said second downstream valve through said clearance.

4. A valve unit as set forth in claim 3, wherein said disc shaped member comprises a first smaller diameter disc and a second larger diameter disc which has essentially the same diameter to said valve member, said second disc being oriented to limit deformation magnitude by contacting with the circumferential edge of said valve member and said first disc being oriented to limit deformation at the intermediate portion of said valve member.

5. A valve unit as set forth in claim 3, wherein said disc shaped member is a resiliently deformable.

6. A valve unit as set forth in claim 3, wherein said disc shaped member comprises a plurality of resiliently deformable discs laminated to each other.

7. A valve unit as set forth in claim 1, which further comprises a plate like member disposed between said first upstream valve and the discharge outlet of said orifice for receiving working fluid flow and distributing the uniform fluid pressure to all effective area of said first upstream valve.

8. A valve unit as set forth in claim 1, wherein said pressure difference between said first and second fluid chamber is created by stroke of said piston assembly.

9. A valve unit as set forth in claim 8, wherein said orifice provides damping characteristics which vary at a rate proportional to two power of the stroke speed of said piston assembly; said first upstream valve provides damping characteristics varying at a rate proportion to two over three power of the stroke speed of said piston assembly; and said second downstream valve provides damping characteristics in response to the stroke speed of said piston assembly lower than or equal to a predetermined piston speed criterion, varying at a rate proportional to two power of the stroke speed of said piston assembly, and in response to the stroke speed of said piston assembly higher than said piston stroke criterion, varying at a rate proportional to two over three power of the stroke speed of said piston assembly.

10. A hydraulic shock absorber disposed between relatively movable first and second objects for absorbing vibration energy which causes relative movement between said first and second objects, comprising:

a cylinder tube defining an internal space and connected to said first object for movement therewith;

a piston assembly disposed within said internal space of said cylinder tube for defining first and second fluid chambers therein, said piston being connected to said second objects by means of a piston rod for movement therewith;

a flow restriction valve unit provided in said piston assembly associated with a fluid path for communication between first and second fluid chambers, said valve unit comprising:

a flow restricting orifice forming a part of said fluid path for permitting fluid flow therethrough at a first limited flow rate;

a first upstream valve associated with said flow restricting orifice and normally closing one end of said orifice for blocking fluid communication between said first and second fluid chambers, and being responsive to fluid pressure difference between said first and second fluid chamber greater magnitude than a predetermined first magnitude to open said one end of said orifice to establish fluid communication between said first and said second fluid path for permitting a fluid flow from said first fluid chamber to said second fluid flow chamber, said first upstream valve including a first resiliently deformable valve member which is normally biased in a direction for closing said one end of said orifice and deformable to increase fluid path area according to increasing fluid pressure difference between said first and second fluid chambers;

a second downstream valve associated with said first upstream valve and arranged downstream of said first upstream valve in series with the latter with respect to the fluid flow from said first fluid chamber to said second fluid chamber, said second downstream valve having a predetermined constant and minimum flow path area of flow restriction path for communication between the downstream of said first upstream valve and said second fluid chamber for providing a predetermined maximum magnitude of fluid flow restriction when the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be smaller than or equal to a second predetermined magnitude, and being responsive to the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be greater than a second predetermined magnitude to shift for increasing said fluid path area;

said second downstream valves being so cooperated with said first upstream valve as to provide linear variation of damping force in accordance with variation speed of fluid pressure difference;

said second downstream valve including a second resiliently deformable valve member which is normally biased in a direction for providing the minimum path area and deformable to increase fluid path area according to increasing fluid pressure difference between the downstream of said first upstream valve and second fluid chambers; and

a stopper means disposed between said first upstream and second downstream valves for defining a maximum deformation magnitude of said first valve member of said first upstream valve means.

11. A hydraulic shock absorber as set forth in claim **10**, wherein said first upstream valve has a valve member capable of shifting in accordance with increasing of the fluid pressure in said first fluid chamber, said valve member being cooperated with a stopper means which limits shifting range of said valve member.

12. A hydraulic shock absorber as set forth in claim **11**, wherein said valve member comprises a resiliently deformable disc, and said stopper means comprises a disc shaped member having substantially the same diameter to that of said valve member so that at least the circumferential edge portion of said valve member seats thereon at a predetermined magnitude of resilient deformation, which stopper means defines a clearance to permit fluid flow from said first upstream valve to said second downstream valve through said clearance.

13. A hydraulic shock absorber as set forth in claim **12**, wherein said disc shaped member comprises a first smaller diameter disc and a second larger diameter disc which has essentially the same diameter to said valve member, said second disc being oriented to limit deformation magnitude by contacting with the circumferential edge of said valve member and said first disc being oriented to limit deformation at the intermediate portion of said valve member.

14. A hydraulic shock absorber as set forth in claim **12**, wherein said disc shaped member is a resiliently deformable.

15. A hydraulic shock absorber as set forth in claim **12**, wherein said disc shaped member comprises a plurality of resiliently deformable discs laminated to each other.

16. A hydraulic shock absorber as set forth in claim **10**, which further comprises a plate like member disposed between said first upstream valve and the discharge outlet of said orifice for receiving working fluid flow and distributing the uniform fluid pressure to all effective area of said first upstream valve.

17. A hydraulic shock absorber as set forth in claim **10**, wherein said pressure difference between said first and second fluid chamber is created by stroke of said piston assembly.

18. A hydraulic shock absorber as set forth in claim **17**, wherein said orifice provides damping characteristics which vary at a rate proportional to two power of the stroke speed of said piston assembly; said first upstream valve provides damping characteristics varying at a rate proportional to two over three power of the stroke speed of said piston assembly; and said second downstream valve provides damping characteristics in response to the stroke speed of said piston assembly lower than or equal to a predetermined piston speed criterion, varying at a rate proportional to two power of the stroke speed of said piston assembly, and in response to the stroke speed of said piston assembly higher than said piston stroke criterion, varying at a rate proportional to two over three power of the stroke speed of said piston assembly.

19. A flow restriction valve unit in a hydraulic shock absorber, which valve unit is provided in one of a piston assembly or a bottom valve assembly and associated with a fluid path for communication between first and second fluid chambers, comprising:

a flow restriction orifice forming a part of said fluid path for permitting fluid flow therethrough at a first limited flow rate;

a first upstream valve associated with said flow restriction orifice and normally closing one end of said orifice for blocking fluid communication between said first and second fluid chambers, and being responsive to fluid pressure difference between said first and second fluid chamber greater magnitude than a predetermined first magnitude to open said one end of said orifice to establish fluid communication between said first and second fluid path for permitting fluid flow from said first fluid chamber to said second fluid flow chamber, said first upstream valve including a first resiliently deformable valve member which is normally biased in a direction for closing said one end of said orifice and deformable to increase fluid path area

according to increasing fluid pressure difference between said first and second fluid chambers;

a second upstream valve associated with said first upstream valve and arranged downstream of said first upstream valve in series with the latter with respect to the fluid flow from said first fluid chamber to said second fluid chamber, said second downstream valve having a predetermined constant and minimum fluid path area of flow restriction path for communication between the downstream of said first upstream valve and said second fluid chamber for providing a predetermined maximum magnitude of fluid flow restriction when the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be smaller than or equal to a second predetermined magnitude to increase said fluid path, and being responsive to the fluid pressure difference between said downstream of said first upstream valve and said second fluid chamber to be greater than a second predetermined magnitude to shift for increasing said fluid path area, said second downstream valve being so cooperated with said first upstream valve as to provide linear variation of damping force in accordance with variation speed of fluid pressure difference, said second downstream valve including a second resiliently deformable valve member which is normally biased in a direction for providing the minimum path area and deformable to increase fluid path area according to increasing fluid pressure difference between the downstream of said first upstream valve and second fluid chambers; and

means provided at an orientation upstream of said first valve member for preventing the fluid flowing through said flow restriction orifice from said first fluid chamber to said second fluid chamber from directly contacting with said first valve member so as to avoid local concentration of the fluid force.

* * * * *